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EXPERIMENTAL STUDY OF SOUND POWER RADIATED FROM COLD MODEL JETS AND GROUND SILENCING ARRANGEMENTS
Volume I

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FOREWORD

This report was prepared by the Acoustics and Seismics Laboratory, Institute of Science and Technology, The University of Michigan, Ann Arbor, Michigan, under Air Force Contract No. AF 33(657)-8802. This contract was initiated under Project No. 8170, "Aerospace Support Techniques," Task No. 817001, "Sound Suppression for Advanced Aircraft and Missiles Bases." This research was assigned The University of Michigan Project No. 05478 and this report is consequently identified as 05478-6-F. This work was administered under the direction of the Support Techniques Branch, Air Force Aero Propulsion Laboratory, Research and Technology Division, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio. Mr. M. Roquemore was technical monitor.

The research program began in December, 1962, and ended in July, 1964. The principal investigator for this research and author of the report was Norman E. Barnett, Research Physicist. Assisting in the research activity were Philip J. Kessel, W. Andres McCaughey, Donald B. Thrasher, and Merle C. Potter. In addition, Professors John H. Enns and Pauline M. Sherman provided technical advice during the beginning phases of the research.

This report is in two volumes. Volume II consists of Appendix III, "Sound Power Data in Tabular Form."

This report was submitted by the author, May 4, 1965.

ABSTRACT

An experimental and exploratory study of radiated sound power has been conducted with cold model jets and various additions to those jets. The objective was to provide insight into the physical acoustical problems related to the design of ground runup silencers for use with advanced jet aircraft. The results demonstrate the complexity of this silencing problem and indicate how the radiated sound power from a given jet can be increased or decreased through several orders of magnitude. The results indicate that various features common to current muffler hardware must act to increase rather than decrease noise with consequent limitation to the silencing obtainable. Further results suggest how to proceed more directly with silencing. A complete exploration of methods and parameters has not been possible and thus additional research is needed to yield all of the insight that can be provided by study of model jets.

This technical documentary report has been reviewed and is approved.

Peter N. Van Schaik

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SECTION 1

INTRODUCTION

The motivation for this research program was the recognized need for a more complete understanding of the physical acoustical problems underlying the design of ground runup silencers for jet aircraft. The broad concept of the existing technical situation regarding ground silencing was expressed in the original USAF request for bid (Reference 1) which culminated in the research reported here. That concept was essentially as follows.

At the inception of this program, a review of existing muffler hardware showed that at best, it was marginally able to silence operational jet engines. Projected larger and more powerful jet engines would soon create even larger amounts of noise and such a trend should be expected to continue. Compensating improvement in muffler effectiveness, along the line of current designs, did not seem likely. However, this projected deficiency in performance of mufflers did not appear to result from any lack of talented engineering development. Rather, it appeared to grow out of a lack of basic knowledge about the physical acoustical mechanisms and parameters underlying the problem area. The existing muffler hardware presented an appearance of "brute-force" adaptation of pre-jet-engine noise control technology.

A review of the relevant jet-noise-research literature revealed a heavy concentration on the nature of noise generation by simple jets, e.g., Lighthill's theory, and noise reduction studies oriented toward the control of noise from jet aircraft during flight. Reference to the ground silencing problem was very meager indeed.

The preliminary study phase of this project reconfirmed the above view. Considered as a whole, both the existing literature and the characteristics of existing muffler hardware led to the conviction that it is necessary to study and to develop much more fully, a basic body of knowledge relating to the ground runup silencing of jet engines. Reference 1 also contained an admonition to concentrate on new and different approaches to the problem and not to merely develop along the lines of existing designs. The research reported here has been guided by the intent of that admonition.

The predominately different requirement between silencing a jet engine during flight and silencing it during ground runup, is the necessity for preserving thrust during flight. During ground or non-flight operations, there is no need to preserve the thrust of the jet-silencer combination. Beyond a tacit requirement that the ground silencer permit normal mechanical and thermodynamic functioning of the engine, almost anything which has a desired acoustic effect might be acceptable. This concept for ground silencing permitted a very wide scope to the research.

The first several months of the program were occupied by a preliminary study which included a specific study of relevant literature, exploratory experiments with improvised jets using the laboratory air line as supply, and an attempt to correlate acoustical, aeronautical, fluid-flow, etc., information and advice into an initial concept for the research program. Apparently also, much of the developmental work conducted by industry has never been fully reported in the literature for a variety of reasons. Thus there is a body of potentially useful facts, observations, etc. which is difficult to obtain or even to find out if it exists. Undoubtedly the present research includes unintentional duplications and deficiencies for want of such information. However that may be, the preliminary study revealed that the state of available relevant knowledge was even more rudimentary than anticipated at the outset. Consequently, it became necessary to start in a most elementary manner and to attempt to identify and isolate the various phenomena which could occur when objects were placed in, around, or in the vicinity of a jet.

In the case of a simple subsonic jet, a very elegant theory of noise generation exists. However, no such comprehensive theory exists for the objects or object configurations to be studied with respect to silencing. In the absence of even vaguely defined guidelines, it became necessary to try all manner of configurations in a sort of "aimed shotgun" approach and from this, to attempt to delineate unique features for more detailed study.

Another revelation of the preliminary studies was the doubtful quality of much of the reported acoustical data. In this research, emphasis has been placed upon obtaining reliable sound-power data within an acoustical laboratory environment. For this purpose, the model jets were operated within a specially-equipped reverberation room in order to provide the desired acoustical data in the most direct and convenient manner possible. Because exploratory experiments may be conducted fortuitously at far from optimum parametrical values with concomitant small acoustical effects, emphasis was placed upon repeatability and high relative accuracy.

The research reported here is by no means complete or exhaustive, and there is a great deal more which can and should be learned from model studies of this type. It is hoped that this research has provided a portion of the knowledge sought. The field of jet noise studies is certainly not static and during the time interval of this program, researches conducted elsewhere have continued to advance the field beyond the state summarized above. Likewise, it is understood that operational mufflers have been much improved in acoustical performance and are close to meeting operational requirements. Nevertheless, a complete understanding of the underlying acoustical phenomena remains an unfulfilled but important goal.

SECTION 2

SUMMARY AND RECOMMENDATIONS FOR FUTURE RESEARCH

An exploratory experimental study has been conducted on the sound power radiated by cold-air model jets when various objects have been placed in and around the jet's exhaust. The primary purpose was to gain a better understanding of the acoustical problems related to the design of ground runup silencers for advanced jet aircraft. The studies emphasized a search for passive configurations which evidenced a potential for silencing by altering in some way the noise generation by a simple high-velocity air jet. The reduced acoustical data have been presented in the form of one-third-octave band sound power levels radiated by the test configuration for a specified mass flow rate of air. Most experiments were performed in the high subsonic velocity range. This research has intentionally disregarded directional effects, the details of the acoustic near-field, and temperature effects in order to concentrate upon the very fundamental property of sound power and to be able to measure this sound power most directly in a reverberation room. The disregarded aspects are not unimportant to full understanding but they constitute complications which are best avoided until the scope of the research has been narrowed and simplified.

The present research has maintained a practical orientation by seeking large acoustical effects, by employing object configurations apparently capable of full-sized realization, and configurations not critical with respect to perfect streamlining. More elaborate schemes for silencing which require auxiliary power or the expenditure of large amounts of material have been avoided as probably impractical for full-size application. Likewise, silencing by the use of ordinary sound absorbing materials was not studied because this method has already been explored extensively by others.

Even with the above limitations, the experiments reported here encompass a wide range and the diversity of interesting results, both positive and negative, are difficult to assemble into a concise but comprehensive summary. However, by omitting the side experiments and concentrating on the ultimate goal of effective silencing during ground runup, several types of results appear prominent.

Experimentally, it seems to be much easier to increase the sound power radiated by a jet than to silence it. The simple jet initially is a comparatively inefficient acoustical source and so the scope for even less efficient acoustic processes appears small. Many types of silencer hardware incorporate noise producing mechanisms as well as noise silencing mechanisms; the net effect being that only a small amount of silencing can be realized.

Of the various objects tested, only screens and metal felts were individually capable of providing large reductions in the radiated sound power. Even these screens and felts had to be selected and located carefully otherwise they were apt to create more noise rather than less.

2.1. SUMMARY ABOUT SCREENS

Screens, as silencing devices, have been partially investigated by others, particularly NACA in full scale tests and also with 4-inch diameter nozzles. (See References 16, 18, 19.) The present research with smaller models agrees with several of the NACA findings, particularly optimization of the silencing effect by using a coarse mesh open screen located about one-half nozzle diameter downstream. The principal reasons for studying screens were to encompass a wider range of parameters, to exploit the higher precision of the laboratory environment, and, by measuring sound power directly, to avoid some of the confusing complications related to directionality. Our experiments indicate larger silencing effects than reported for full-scale tests (Reference 18) and even somewhat larger than for the NACA model experiments (Reference 19). The exact reasons for these differences of magnitude have not been resolved because our experiments and the NACA experiments differed in too many details to allow a full comparison of results.

Our research on screens demonstrates that the silencing effect, particularly its magnitude, depends markedly upon the initial velocity of the jet; at least within the high subsonic range. More pronounced silencing was invariably obtained for the higher velocity test condition (a fortunate attribute for practical silencing). For a given screen, a distinctive pattern of spectral shifts was produced by varying the distance between nozzle and screen. Generally the spectral changes produced by changes in the configuration parameters were large and complex enough to frustrate a meaningful description in terms of broadband sound alone.

Screens were capable of providing silencing down to the lowest frequencies measured, a feature which does not seem to have been demonstrated clearly in previous research. The more open screens gave more silencing, especially toward the higher frequencies, but the parameter of percentage open area has been only partially investigated. The limited results suggest that future research might concentrate on screens which are 40% or more open. Screens also displayed a tendency toward a silencing deficiency at or near the upper frequency limit of the measurements; this effect became more pronounced as the initial jet velocity was reduced. This result poses a problem for future study and also suggests that screens may need to be combined with some other silencing mechanism to create a simple broadband silencer. In this respect, conventional sound absorbing materials might provide the complimentary frequency-characteristics. The combination of absorption and a screen described in Reference 19 represents one example of such a composite design but there are many different arrangements which should be tried. The fact that the

screen may be placed at an angle across the exhaust flow holds the possibility of deflecting the flow and simultaneously achieving useful silencing.

This research demonstrates that two screens, properly selected and arranged in series, can yield more silencing than the best single screen. More research is needed to fully delineate the potentialities and the limitations of arranging screens in cascade. It looks as if two screens in series can provide worthwhile acoustical benefit but that more screens are of doubtful value.

Several tests demonstrated, using models and the reverberation-room sound power measurement technique, that the silencing effect of a single rod or wire stretched across a jet could be measured all across the spectrum. This result opens the door to a much more basic area of investigation. It appears definitely possible to experimentally investigate the acoustical consequences of the size, shape, and position of a single obstructing object placed across the flow. Such an investigation could encompass the entire frequency spectrum not just aeolian tone generation. The simple geometry of the test configuration would permit the meaningful application of flow visualization techniques. Likewise, the simple geometry makes concomitant analytical and theoretical investigation much more attractive than say for the case of a complete screen.

2.2. SUMMARY ABOUT METAL FELTS

The experiments using sintered metal felts demonstrated an even larger potential for silencing than did the experiments with screens. In many respects, however, the acoustical behavior of felts resembled that for screens. For example, felts gave better silencing when placed rather close to the nozzle and the changes in spectral distribution as a function of configuration parameters followed complicated but recognizable patterns. Most of the felt experiments were performed with a 5% dense material of one nominal fiber size; a limitation imposed by readily available material types. A 20% dense felt of the same fiber size gave much less silencing but the parameters of density and fiber size need a much more complete investigation.

The effectiveness of different thicknesses of 5% dense felt was investigated over a range from 1/8" to 1" and the thinnest layer found best. This result leaves the thickness range from 1/8" down to a "single-layer random-mesh screen" to be investigated. The better of the metal felt configurations do not seem to leave as much residual high-frequency noise problem (or else they displace it to even higher frequencies). The metal felt experiments indicate silencing down to the lowest test frequencies in a magnitude which often exceeds that found for the best screens. Although several plausible explanations for the acoustic behavior of these metal felts come to mind, it is not yet clear how a felt achieves its silencing effect or why it is more effective than a screen.

All features considered, a single layer of metal felt is promising for development into a very simple light-weight silencer. It can yield much more silencing than a single screen. If these results with cold model jets hold true for full-size hot jets, a single layer of metal felt would be almost as effective as some existing runup silencers. For example, Figure 59 displays a broadband silencing of 26.5 db achieved with a single layer of metal felt.

Experiments indicate that metal felts can be supported by screens, to assist in resisting the exhaust flow forces, without drastically altering the acoustical behavior. In some combinations, a slight improvement over the metal felt alone was experienced. There was also indication that a coarse screen placed ahead of the metal felt could produce small but useful acoustical consequences. If the observed improvements were not the consequence of incompletely optimized metal-felt parameters, then these combination effects should be exploited.

2.3. SUMMARY ABOUT MODEL SILENCERS

In another line of investigation some experiments were conducted with model structures which resembled some existing runup silencers. These models were L-shaped configurations of tubing which did not attempt to model the finer details nor the internal structure of any existing silencer. Tests with these model configurations evidenced much enhanced low-frequency radiation as well as the specific tonal characteristics of acoustical pipes. Qualitatively, these results were to be expected from general acoustical knowledge but more interesting is the information about the magnitudes of the sound power associated with these results, which magnitudes are not ordinarily predictable from general knowledge.

Additional experiments with a perforated model, a perforated model wrapped with fiber glass, and a water-inflated flexible model all reinforce the contention that solid walls surrounding a jet exhaust will enhance the amount of low-frequency sound power radiated from the configuration. Mechanical resonances and coincidence effects in the walls of these models do not appear to have played any significant role in these experiments but, if such effects did occur in a silencer, they could only decrease the amount of silencing.

The implications from the model muffler-body experiments are that the very existance of solid walls must act to increase the radiated power and that their geometrical configuration will determine the superimposed tonal characteristic; both completely undesirable consequences. Therefore most existing runup silencers would seem to start with a large step in the wrong acoustical direction but then recover by an even larger step in the right direction which is a consequence of their internal silencing mechanisms, mainly absorption probably. It would appear more efficacious to achieve

silencing by taking only steps in the right direction if possible.

2.4. SUMMARY ABOUT NEW DESIGNS

From this research, one concept of silencer design developed which required that extended solid surfaces be completely avoided. In this way, both the broadband enhancement and the normal mode effects might be largely avoided. One test configuration based on this concept was fabricated entirely from metal felts. The body parts were cut from thick layers of felt and surrounded the exhaust flow and only a thin layer of felt intercepted the flow. (See Figures 70 and II.25) This configuration yielded a large and consistent silencing effect across the entire experimental frequency range and, for an initial jet velocity of Mach one, demonstrated a broadband sound power reduction of 27.8 db. It is reasonable to assume that this experimental configuration was not fully optimized with respect to silencing or compactness but even in this form it represents a compact, light-weight silencer. Moreover, many variations to this configuration are possible, all adhering to the original concept, and further experiments may prove some of them to be distinctly more advantageous.

A different experimental configuration was devised using metal felts and screens in various series combinations. A solid outer boundary, in the form of a brass tube, was used to support this configuration and, while the brass tube violates the concept of avoiding solid boundaries, the thick lining of metal felts seems to have at least partially isolated (acoustically) the solid boundary from the jet. (Probably, when it is not in the high velocity air flow, a metal felt functions as an ordinary porous acoustical absorbing material. However, in the present case, the unusual ratios of physical dimensions to wavelength make quantitative predictions of effectiveness very uncertain.) The best of these configurations yielded a 38.9 db reduction in the radiated sound power (See Figures 73 and II.26C). Again it is reasonable to assume the further development might lead to more silencing and a more compact design.

A variety of untried ideas stem from the various metal felt and screen configurations. One such idea would employ a continuously distributed felt-like material to constitute the complete silencer. The density distribution and other parameters could be varied as necessary to achieve silencing and at the same time, the exhaust flow might be diffused and directed by virtue of the varied flow resistance.

A further line of investigation dealt with the combination of a model muffler-body with screens and metal felts. In most cases, the results are approximately those one would expect from a linear combination of the noise enhancing effects of the muffler body and the noise reducing effects of the screens and felts. In one case, however, a very much larger noise reduction was obtained. (See Figure 86). Possibly this configuration functions as a

reactive type of acoustical filter although the experimental conditions are far removed from those usually selected when one is consciously attempting to apply acoustical filter theory. The size of the acoustical result merits further investigation although this configuration, as it now stands, would represents a large and heavy silencer. An absorptive lining would attenuate the residual high-frequency noise. It may be that existing runup silencers already utilize the phenomenon discussed above, but if not, some practical application of it may be desirable. It is especially interesting that the band of large attenuation appears to start on the low-frequency side at the frequency of the lowest-order pipe mode.

2.5. SUMMARY OF SILENCER CONCEPTS

The initial concept of the runup silencing problem based on the noise from simple subsonic jets of equal mass flow rates, discussed in Section 3.1, still remains useful in spite of experimental difficulty in fully realizing its predictions.[1] However, after performing the reported research, a somewhat broader concept, which can be argued somewhat as follows, is appealing. A jet exhaust consists of a specifiable mass flow-rate of hot gas which in turn can be specified in terms of chemical composition and an appropriate equation of state. The energy of the jet exhaust consists of thermal energy (represented by the random motion of the gas molecules) appropriate to the gas temperature and the kinetic energy of the ordered fluid flow which is directed principally along the jet axis. The ambient air is at a much lower temperature (its mean random molecular velocity is much lower than for the exhaust gases) and it has little or no ordered dc flow except for winds. When the jet exhaust gases have mixed with the surrounding air, the mixture is characterized approximately by a small increase in temperature above the ambient value and a small ordered (radially outward) flow.

The random molecular motion of the ambient air corresponds to an irreducible background of acoustical noise which at normal temperatures lies 10 to 20 db below the threshold of human hearing. The intensity of such thermal noise depends explicitly on the first power of the absolute temperature of the gas (to good approximations) so that all other factors being equal, a gas would have to be exceedingly hot for its inherent thermal acoustic noise to be intense and therefore subjectively loud. Indeed upon leaving the nozzle, if all of the ordered flow energy in the jet exhaust could be instantaneously transformed into random molecular motion, constituting a gas cloud at an even higher temperature, the acoustical noise of thermal origin in such a cloud would still lie near the threshold of hearing.

[1] The several concepts of runup silencing discussed here have not been found explicitly expressed in the literature reviewed by the author but the underlying physical principals are so widely known that any aspect of originality seems doubtful.

The ground runup silencing problem appears to be how to get from the ordered dc flow at the nozzle to completely random molecular motion without exciting other noise generating mechanisms along the way. The normal noise from simple jets represents a natural mixing process which generates an appreciable amount of noise during the transition from one gas state to the other. The concept of runup silencing used in Section 3.1 was contained within the concept of simple jet noise inasmuch as it started and ended with an ordered dc flow. Thus it appears to constitute just one special case in a much more general framework.

Taking a different path, the flow from a jet might be "diffused," perhaps by means of a controlled spatial distribution of flow resistance, so that the ordered exhaust flow becomes diverged into a much larger solid angle than the roughly 0.007 π solid angle of a simple jet. The more rapid divergence would quickly reduce the mean outward flow velocity to low values and considerably alter the distribution of shear forces which account for the noise from simple jets.

Another possibility might be to break up the initially ordered dc flow into essentially random motion by means of scattering obstacles placed in the exhaust flow (somewhat similar to the wind-tree model used in the kinetic theory of gases). The scatterers would need to be arranged in a three-dimensional spatial distribution and, individually, they should present only small areas of solid surface so as not to enhance the radiated acoustic power in the frequency range under consideration. It seems likely the silencing observed for screens and metal felts is related to these concepts of divergence and scattering.

2.6. SPECIFIC RECOMMENDATIONS FOR FUTURE RESEARCH

With respect to future research, there are a variety of investigations which ought to be pursued to take fuller advantage of model studies with respect to jet noise and runup silencing.

A. The reverberation-room (containing a rotating vane) method for measuring sound power has been demonstrated to be a very fruitful method for conducting model scale research on jet noise and jet silencing. It should be utilized as a basis for more research on jet noise both basic and applied in terms of ultimate objectives.

   Experience from the present research indicates that somewhat larger air flow rates through a reverberation room can be tolerated without violating the necessary acoustical conditions within the room. Thus future research of this type can employ somewhat larger model nozzles to advantage, not the least of which would be to shift the noise spectra downward to fall more nearly in the middle of the measurable frequency range. The air flow control system is reasonably satisfactory in the present form but the acoustically and aerodynamical

clumsy calming chamber should be replaced by a silencer similar to Figure II.26. It should be more effective acoustically and such a design could leave the nozzle end available for particularly refined streamlining intended to minimize upstream flow separation at all flow rates.

Essentially this recommendation is, now that this particular experimental method has been proven advantageous, learn from it and exploit it fully as a very powerful tool in several areas of silencer research. Previously, this tool has been almost totally neglected by researchers in jet noise and much of the present research program had to be devoted to learning how to apply this tool to this particular problem area.

B. Extend the research on screens, metal felts, distributed flow resistance, flow scatterers, etc., with the ideas both (1) of finding empirically the most effective materials and configurations and (2), in conjunction with theoretical and analytical work, of learning the detailed physical processes involved. Correlative to such research should be the performance of several selected experiments at both model and full-scale to investigate such characteristics as scaling factors, directionality, and temperature characteristics. Certain conjunctive experiments at model scale conducted in anechoic surroundings would be in order also. The correlative group of experiments would almost certainly require the participation of several laboratories.

C. Develop further the experimental silencer configurations which employ metal felts without solid boundaries and the metal felt-screen combinations similar to those shown in Figures II.25 and II.26. Some preliminary development should be carried out at model scale to optimize performance and compactness. This should be followed by full-scale development into experimental production prototypes of one or more versions of such silencers. Extrapolating from the present research at model scale, many advantages are anticipated with respect to size, weight, and performance.

D. Follow up the exploratory measurements of the sound power consequences of single wires stretched across a model jet. Take advantage of the simple geometry to concurrently apply the aerodynamicist's flow visualization techniques. The fact that the reverberation-room measurements used in this research permit such accurate and controlled experiments provides an exciting possibility of bridging a gap which now exists between the fields of aerodynamics and acoustical physics and their respective experimental methodology.

E. Continue the exploratory research into more and "new" methods of silencing jets at model scale. At the start of this program, the

general state of knowledge about the problem area, as reflected by the literature, was diffuse and fragmentary. Consequently, after demonstrating the applicability and advantages of model experiments in the reverberation room, the research program had to be almost of the shotgun type and, even after appreciable accomplishment, much more investigation remains to be done. Most of the existing general acoustical knowledge about noise sources relates to the expected frequencies or spectral distribution. Relatively little is known quantitatively about the associated sound power on either an analytical or empirical basis; simple unsilenced jets constitute the principal exception to this statement. Therefore, particularly with respect to silencing, much research will be needed to provide the full basic knowledge essential to the routine engineering design of practical silencers. Considering the relatively low cost of model studies, such as reported here, in comparison with many full-scale experiments and their associated facilities and hardware, continued model-scale studies would seem to constitute a good investment toward future knowledge.

SECTION 3

INITIAL CONSIDERATIONS

The concepts and considerations governing the reported research are outlined in this section. The underlying ideas are certainly familiar to acoustical scientists but some of these ideas would seem to be less familiar to many engineers working with jet engine noise as evidenced by the general literature on the subject. Moreover, the use of a reverberation room as an instrument for the measurement of radiated sound power is less widely known than is the use of an anechoic room. This discussion is somewhat more comprehensive than usual in a research report in order to present clearly what was done and why it was done.

A first question to be answered was how to go about studying the acoustics of ground runup silencing. It seemed, after consideration of the general situation described in Section 1, that experimental model studies were indicated. The experiments to be undertaken would have led to an excessively expensive, technically difficult,and very laborious program if they had been performed with a full-sized operational jet engine. Moreover, the ability to collect sufficiently precise acoustical data to detect small trends is doubtful in an out-of-doors, free-field acoustical environment. By contrast, model studies conducted within the confines of an acoustical laboratory environment seemed quite feasible. Of course, due consideration must be given to scaling factors and other deviations from the real, full-sized jet engine situation when interpreting the results.

Nevertheless, the essential broad-scope studies appeared to be feasible only if conducted in the laboratory. And moreover, it seemed unlikely that one could make much progress in the silencing of real jets unless he was able to cope with simple, cold model jets. Then transition from model to real jets could be undertaken at a later date by means of a few simpler and carefully planned decisive experiments. With limitation of the research program to model studies, the research became appropriate for a university acoustical laboratory.

3.1. CONCEPT

As a result of preliminary considerations, it was decided to base this research on the measurement of total radiated sound power for a controlled mass flow of air through a specified nozzle representing the jet engine. The mass flux of fluid from a particular nozzle area is directly related to the mechanical energy of the jet and we can arrange to directly evaluate the mass flow of air supplied to the nozzle configuration under study. The total radiated sound power represents that portion of the jet's mechanical energy

which becomes transformed into the undesired form of acoustical power. Experiments so oriented appear to attack the silencing problem most directly. Those muffler arrangements which yield the least sound power for a given mechanical energy would be considered best.

The concept of radiated sound power emphasizes the role of the acoustic far-field and total power de-emphasizes the role of the spectral distribution of the sound. Of course, it is not reasonable to completely neglect the spectral distribution and indeed, the nature of acoustical measurements demands some determination of the spectrum since total power is a quantity calculated by integration across a spectrum. However, in a fundamental investigation, first comparisons based on total power regardless of spectral distribution are valid. It is only when one is studying more specific applications that a more powerful but preferred-spectrum sound might be acceptable. Also, as it turns out, one can not actually evaluate total acoustic power unless he possesses knowledge about the entire spectrum. Acoustical instrumentation imposes a finite bandwidth upon the measurements and so total sound power necessarily becomes replaced by sound power contained within a broad but still finite bandwidth.

Emphasis upon the acoustic far-field is also justifiable. True, real-life silencer applications include the possible placement of men and structures within the acoustical nearfield of mufflers. However, the research planned is of broad scope and permits the investigation of the acoustical performance of all kinds of arrangements to the jet. The detailed structures of the noise sources thus created can be almost infinitely varied and complicated. Thus the only feasible approach is to rank order on the basis of far-field performance and then later, if necessary, select among a more limited variety of muffler configurations on the basis of near-field noise characteristics. Physically, the prediction of far-field sound power from some, but incomplete, knowledge of the nearfield characteristics of an arbitrarily complex source is essentially impossible. Even under the most favorable conditions, prediction on the basis of near-field information is orders of magnitude more difficult than direct measurement of the far-field. Experience derived from a broad gamut of noise reduction research emphasizes the practicality of following the procedures argued above.

Directionality of the acoustic fields also deserves consideration. Practically, total radiated sound power can be evaluated only in a free-field or else in a completely diffuse field. (Intermediate field arrangements require a more complete and detailed knowledge of all boundary conditions than is usually available.) By its nature, a diffuse field can only provide information about source output integrated with respect to direction. The diffuse-field measurement of sound power was selected for this program to take advantage of the simplification which the inherent integration with respect to direction provides. Every configuration to be tested potentially could have a different directionality characteristic at every frequency so that a free-field measurement approach to evaluation of

total sound power would be excessively laborious. Furthermore, regarding the practical application of mufflers, very pronounced direction characteristics generally would not be realizable because of real gas characteristics in contrast with ideal gas characteristics, atmospheric turbulence, thermal effects, etc. In this program, we wish to concentrate upon obtaining orders of magnitude reduction in radiated sound power and just as in the case of near-field effects, consideration of directionality effects must be deferred until much later.[2]

Now consider the jet engine's operating conditions which are apt to prevail during ground runup operation. The exhaust velocity can range from low subsonic values during starting and idling operations to somewhere in the vicinity of sonic velocity at full-throttle operation. Without ram air pressure, the simple jet engine is not apt to exceed critical pressure ratio by very much.[3] Consequently, most ground runup operation produces a subsonic or, at most, a slightly supersonic exhaust stream. Furthermore, mufflers will most likely have a subsonic efflux of exhaust gases. Initially, it seems appropriate to concentrate entirely upon learning how to silence subsonic jets.

Supersonic or overchoked jets contribute an additional noise generating mechanism in the form of shock fronts and hence would tend to complicate the research program. Other research (see, for example, Figures 6a and 3a, Reference 3) has indicated that the exhaust flow may become slightly supersonic, say about Mach 1.05, before the presence of an additional noise source reveals itself as an excessive increase in acoustic power.

It was elected to concentrate these studies on silencing of subsonic jets because:

(1) During ground runup, operational jet engines ordinarily would be, at most, only slightly supersonic.

[2]The general philosophy of acoustical research measurements, on which the above remarks are based, can be formulated rigorously from the physics of acoustical fields and sources. As such, most of the fundamental ideas are contained in all textbooks on classical acoustical physics. However, those ideas are seldom organized with the intent of guiding an experimental research program and they are usually not tempered by the practical restrictions imposed by current acoustical instrumentation. As a consequence, it is not possible to refer to one or two pertinent references for support. A reasonably complete and satisfactory development of this topic has only been found possible so far within the content of a two-credit-hour graduate-level college course. (Reference 2)

[3]After-burning and other thrust augmentation schemes may cause modest increases beyond critical pressure ratio.

(2) The efflux from any muffler will very probably be at low subsonic velocities.

(3) The noise generation by subsonic jets is more fully described in the literature allowing firmer establishment of our point of departure.

(4) At the very least, an effective muffler must be effective against the noise from subsonic jets.

An operational jet engine potentially can generate and radiate noise in a variety of ways in addition to noise from its exhaust stream. However, these other noise sources present rather conventional noise reduction problems and probably can be treated in a straightforward manner. In any event, the jet exhaust noise is overwhelmingly more powerful and must be silenced first. It is also unique as a noise reduction problem because the noise originates in space outside of the machine which ultimately causes it. Moreover, the region of noise generation is physically large. Thus the conventional noise control approach of enclosing the source, attenuating, and absorbing the sound energy would lead to an unacceptably monstrous muffler arrangement. Rather, it would seem necessary to prevent the generation of noise by the exhaust stream or to drastically reduce the efficiency of noise generation by it without introducing other prominent noise generation mechanisms.

The noise produced by a subsonic jet has been studied extensively by many researchers and the related literature is voluminous. References 4 and 5 provide useful summaries of the existing knowledge. Moreover, a fairly satisfactory theory describing the generation of acoustic power by simple subsonic jets has been developed by Lighthill (see Reference 6) and it has been augmented and elaborated by many others. (For example, see References 7 and 8.) Considered as a whole, experiment and theory have provided a fairly comprehensive description of the noise in terms of total acoustic power, directional pattern, and frequency distribution. This description provides a starting point for the present research.

Figure 1 illustrates schematically the cross section of a simple subsonic jet. In this illustration, d is the diameter of the nozzle exit and x is the distance measured downstream from the nozzle exit. In some of the later discussions, it will be convenient to utilize the dimensionless ratio x/d to specify downstream locations.

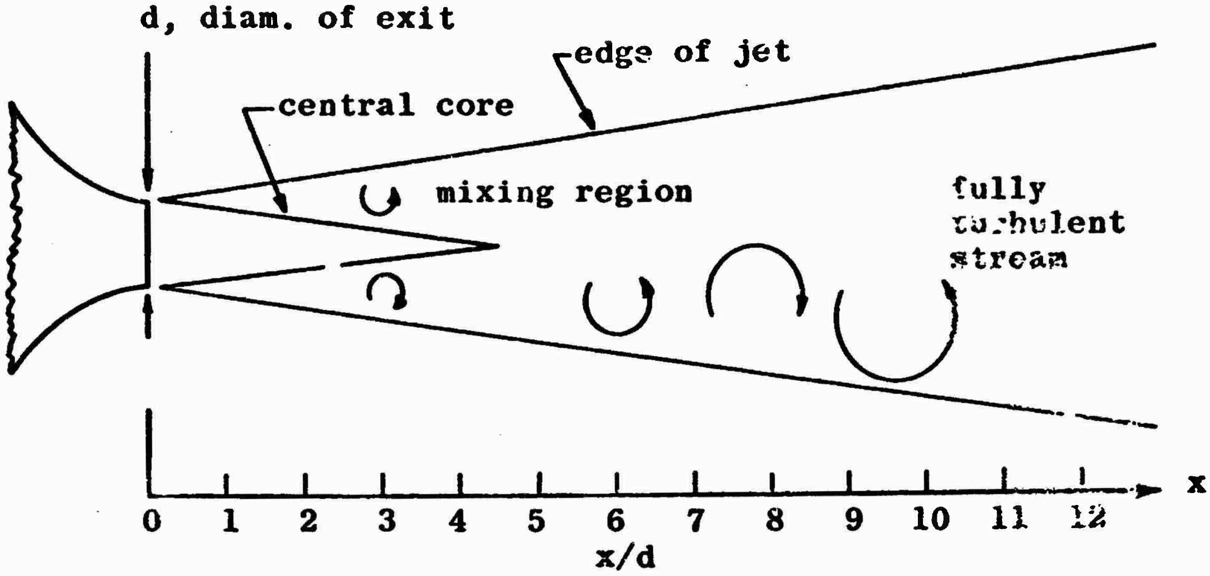

FIGURE 1. SCHEMATIC SECTION OF SUBSONIC JET EXHAUST

For present purposes, it will be sufficient to describe the total acoustical power generated by a stationary cold ambient-air jet as: (see Reference 4)

$$W_A = K \rho_o \frac{U^8}{c_o^5} d^2 = \eta W_M \qquad (1)$$

where W_A = total radiated sound power, watts

W_M = mechanical power in the jet, watts

η = acoustical efficiency

K = acoustical power coefficient

ρ_o = density of the ambient atmosphere

U = mean velocity of the jet flow

c_o = velocity of sound in the ambient air

d = diameter of the nozzle exit.

Typically, K has a value in the vicinity of 6×10^{-5} while η is of the order of $(U/c_o)^5 \times 10^{-4}$. For a specified ambient air condition, the essential relation described by Equation (1) can be expressed as:

$$W_A \propto AU^8 \qquad (2)$$

where

$$A = \text{area of nozzle exit} = \frac{\pi}{4} d^2$$

The placement of the frequency of maximum sound power within the spectrum is given by:

$$f_m = S_m \frac{U}{d} \qquad (3)$$

where

f_m = frequency of maximum sound power

S_m = Strouhal number corresponding to f_m (roughly a constant of order 0.2)

The sound power spectrum is distributed as illustrated in Figure 2 which expresses the relationship between dimensionless frequency ratio f/f_m and the power level of a band in db below the total sound power level. The highest

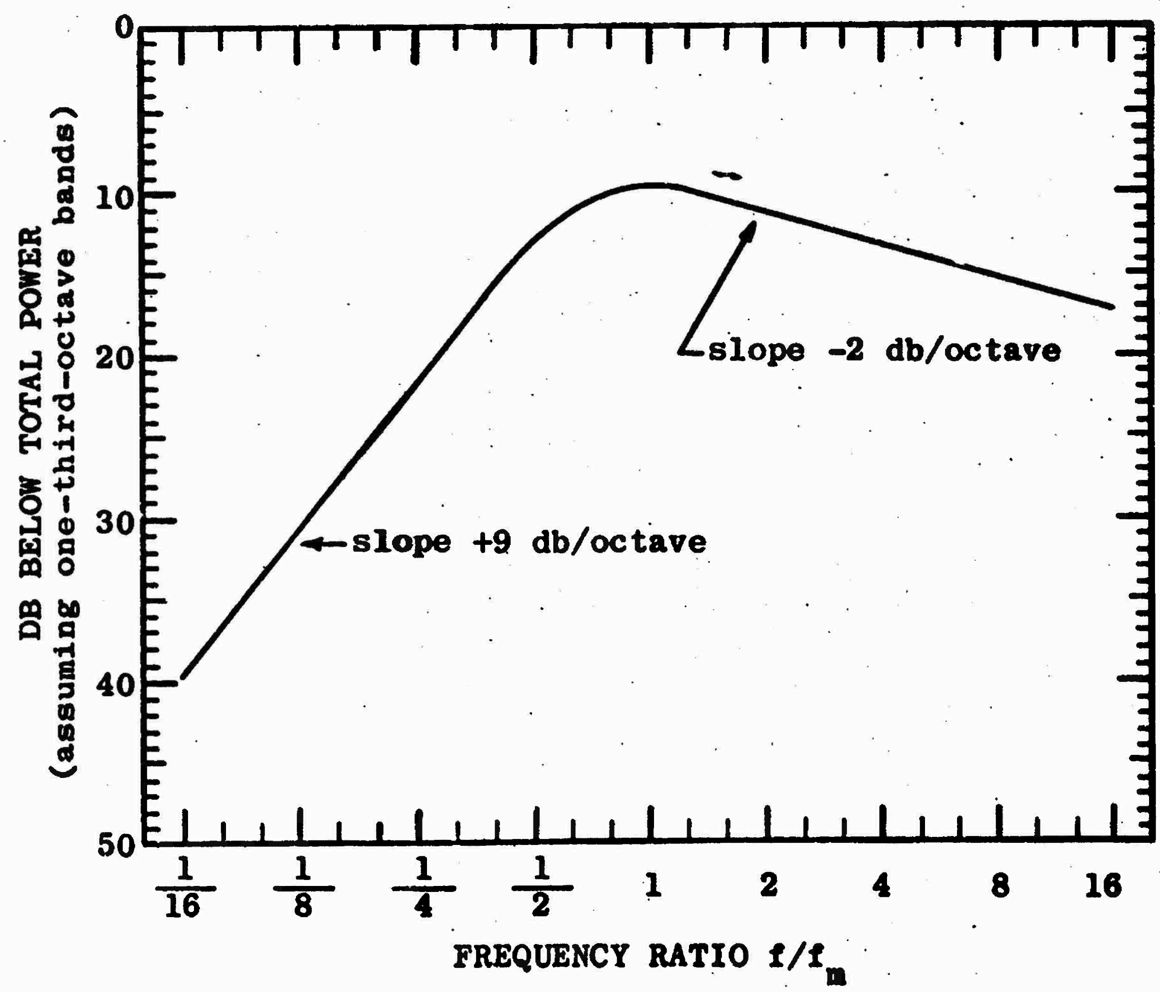

FIGURE 2. GENERALIZED SPECTRUM FOR AN AMBIENT AIR JET

frequency noise originates close to the nozzle exit, that is, at small values of x or x/d while lower frequencies are generated at correspondingly larger distances downstream. (See Reference 5)

This general description of subsonic jet noise has been verified for jet diameters ranging from a fraction of an inch to more than twenty inches. Therefore, this description may be accepted as providing the given point of departure for studying jet silencing and it establishes the reference sound power against which muffler effectiveness is to be measured.

In simplest physical terms, one starts with a given mass flow of air which will be accelerated by rapid contraction in a nozzle and then will be released into the surrounding still air as a high-velocity stream of gas, initially devoid of turbulence. The contraction occurs rapidly enough so that the boundary-layer at the nozzle exit cannot have grown very thick. Then, as a result of the large amount of shear existing between the high-velocity gas stream and the quiescent ambient air, turbulence and, consequently, noise are generated. The growth and distribution of this turbulence are the natural result of the dynamic forces and the physical characteristics of the real gases involved. Such is the fundamental nature of the noise source which this research seeks to learn how to control.

All other physical characteristics such as high temperature in the jet, composition of jet fluid differing from the surrounding atmosphere, turbulence or combustion noise existing upstream of the nozzle exit, etc., are at most of secondary importance to this study. And, in principle, many such characteristics could be taken into account in a properly formulated equation of state for a real gas.

The silencing problem may be visualized somewhat as shown in Figure 3, where the subscript 1 refers to the parameters representing the unsilenced jet and the subscript 2 refers to the parameters describing the outlet of the muffler. To the extent that the "Lighthill" theory correctly describes turbulence noise, the situation depicted in Figure 3 represents the limiting case of maximum possible silencing. It is assumed that no additional noise is generated within, or because of, the muffler; or, at least, that any such noise is dissipated before it reaches the muffler's exit. It further tacitly assumes that the "Lighthill" theory remains a substantially correct description of the noise generated by the muffler's efflux. However, the detailed operation of the muffler can involve any mechanisms imaginable such as alteration of the normal formation and growth of the turbulence, sound absorption, sound cancellation, alteration of noise generating efficiency, etc. Indeed, the rectangular box representing a muffler in Figure 3 may itself be conceptually misleading for at this point, there is no intention of implying any structural form to a muffler. It is only assumed that after silencing, there will again be a noise generating mechanism in the form of a simple jet exhaust, external to any muffler arrangement but producing less sound power because of a larger effective area and

a lower effective velocity. If the "Lighthill" theory should be inappropriate to very low-velocity effluxes, then some more suitable description must be substituted to define the limiting case for large amounts of silencing.

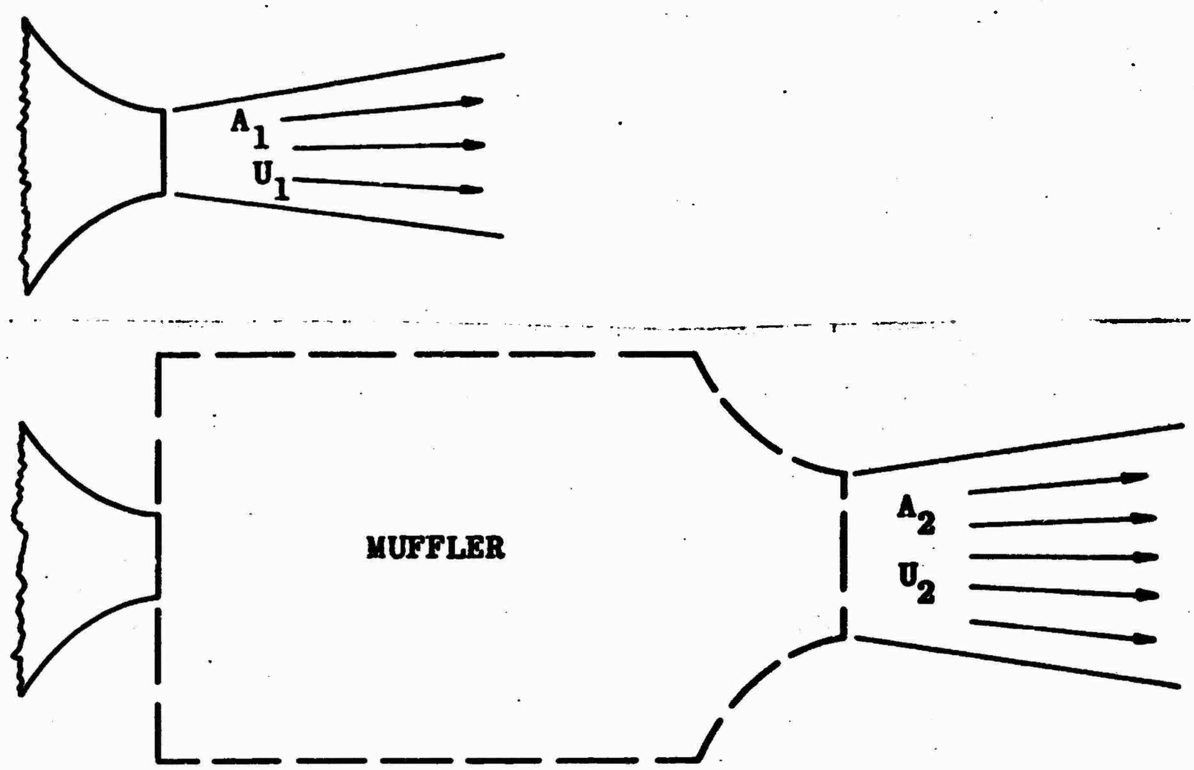

FIGURE 3. SCHEMATIC REPRESENTATION OF SILENCING

Continuing the argument, if the noise remaining after silencing is greater than assumed above, then the muffler has not achieved perfect performance and is itself contributing excessive noise. The research approach to be utilized here is to arrange a high-velocity model jet corresponding to an unmuffled jet engine and to see if the predicted sound power and spectral distribution can be verified. Next, arrange a larger but lower-velocity model jet corresponding to the efflux conditions of some hypothetical muffler and again observe if the anticipated sound power and spectral distribution are obtained. The observed acoustical differences represent the limiting results to be expected from a perfect muffler. Finally, one returns to the high-velocity jet and attempts, by various silencing stratagems, to achieve the acoustical performance typified by the lower-velocity jet.

It might be asked, "Why not utilize muffler designs similar to those applied to automobiles because, after all, an extensive body of research and engineering practice already exist for such mufflers?" (See, for example, Chapter 21, Reference 4.) There are several reasons why such muffler theory and practice are not applicable to jet noise problems; at least, not in any direct manner. The noise to be quieted by an automobile muffler is generated back in the engine and conducted into the muffler by an exhaust

pipe. The equivalent arrangement for a jet engine would possess titanic proportions. Also, conventional muffler theory is restricted generally to arrangements in which the transverse dimensions of the component parts (tubes, chambers, etc.) are much less than the wavelength of the sound being filtered. The physical dimensions and frequencies involved in jet exhaust noise violate such a restriction to an extreme degree.

3.2. EXPERIMENTAL PROCEDURE

In order to advance from the concept developed above, it is necessary to determine whether or not the sound power and the spectral distributions from the assumed model jets are compatible with available acoustical facilities, principally the reverberation room. In a sense, this consideration involves choosing the scale size for the experiments.

Preliminary calculations indicated that 100 SCFM (standard cubic feet of air per minute) passed through a one-half-inch diameter nozzle would just reach sonic velocity at room temperature, about 1130 feet per second. Under these conditions and assuming a maximum Strouhal number of 0.3 (somewhat conservative since most references suggest a value nearer 0.2 for small model jets), the peak frequency, f_m, expected according to Equation (3) would be about 8000 cps. A smaller Strouhal number would lower this frequency. Similarly, larger diameter nozzles and lower flow velocities can be expected to produce lower values of f_m. Since our reverberation room was known to perform well up to at least 10,000 cps, it should be possible to measure the acoustic output from subsonic jets as small as one-half inch in diameter from a standpoint of frequency range. Reference 9 substantiates this contention for it reports satisfactory sound-power measurements by the reverberation room method on small jets of similar size.

A calculation of expected total sound power according to Equation (1) for a one-half-inch sonic jet results in about 0.35 watts or 115.5 db with respect to one microwatt reference power. From previous experience, this amount of power would generate a sound pressure level of about 110 db in the reverberation room. While such a sound pressure level is uncomfortable to the unprotected ears, it would not cause any physical difficulties in the reverberation room. Special high-intensity microphones would not be needed. Still, it would be sufficiently high so that even after considerable silencing, the residual sound could be measured readily and accurately with respect to level.

Figure 4 illustrates the noise spectrum to be expected from a one-half-inch diameter nozzle operated at an air flow of 100 SCFM. This is the operating condition chosen to simulate the unmuffled jet engine during ground runup. Figure 4 also includes predicted spectra for the same mass flow of 100 SCFM through nozzles 0.707 and 1.000 inch in diameter (twice and four times the area) respectively.

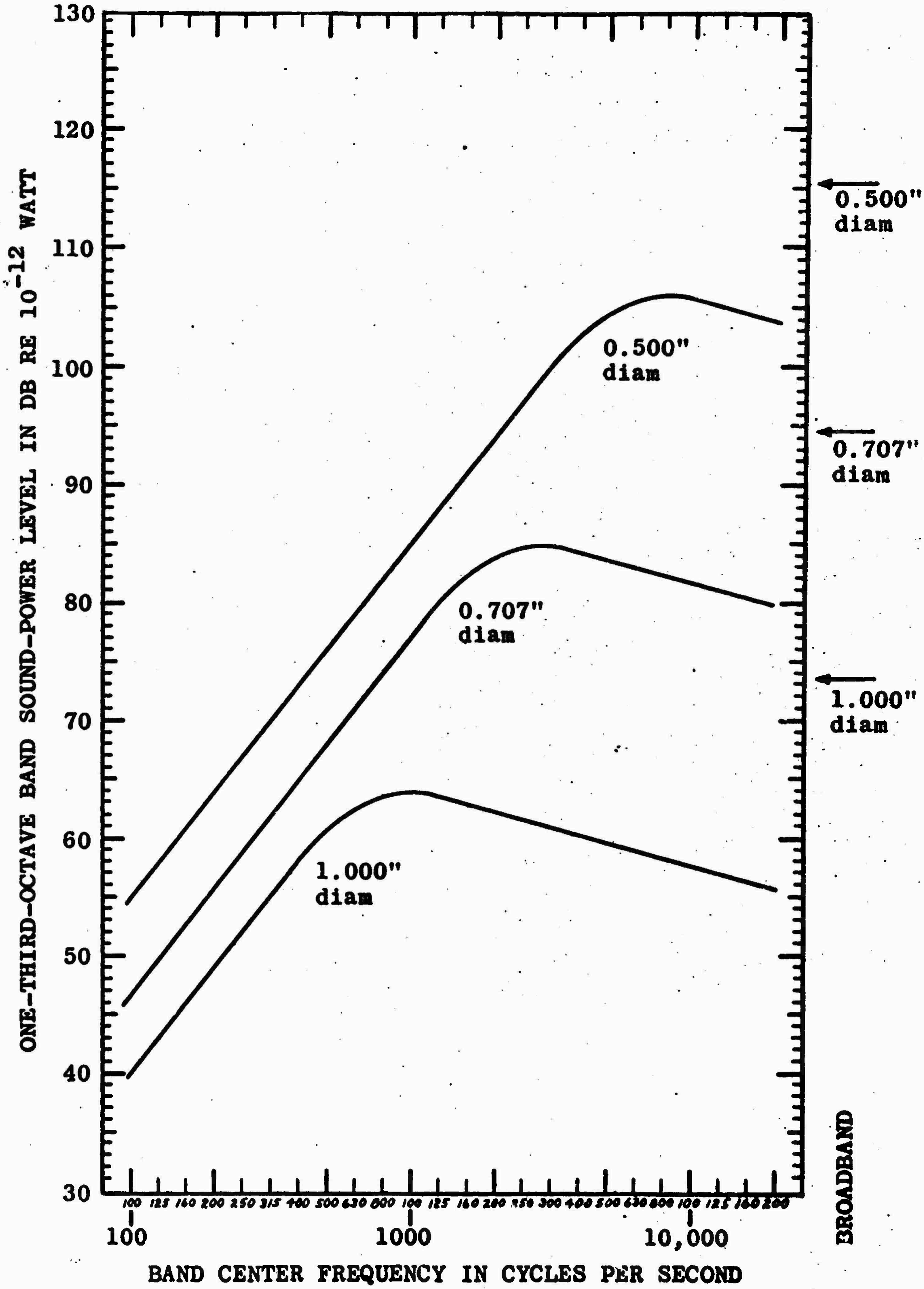

FIGURE 4. EXPECTED SOUND POWER SPECTRA FOR SIMPLE NOZZLES OPERATED AT 100 SCFM.

Neglecting temperature and density effects, doubling the exit area would halve the velocity for the same mass flow and application of Equation (2) indicates that the acoustic power would be reduced in ratio 2^{-7} or -21 db. Likewise, application of Equation (3) indicates that f_m would be shifted downward in ratio $2^{-3/2}$. These ratios have been used in constructing Figure 4 and the resulting lower curves are representative of the noise which might be anticipated if the corresponding perfect mufflers can be devised. The curves in Figure 4 also verify that the anticipated experimental research results will fall within an appropriate frequency and magnitude range, permitting accurate measurement with normal acoustical instrumentation.

The instrumentation actually employed is described in Appendix I entitled Experimental Facilities which should be consulted for details. It was decided, however, that one-third-octave band measurements would provide sufficient spectral detail. Narrower band measurements could have been performed but they were considered unessential and unnecessarily laborious for the present program. One-third-octave band measurements generally provide sufficient resolution to follow gradual changes in the spectrum shape of continuous spectrum sounds and also to indicate the presence of strong pure tones by abrupt changes in level between neighboring bands. Since mufflers causing the generation of strong pure tones probably would not be acceptable silencers, more detailed investigations of suspected pure-tone spectra would not be needed in this particular program.

Acoustical power measurements using a reverberation room require the observation of the space- and time-averaged sound pressure level due to the noise source and the observation of the decay rate in each frequency band. A microphone possessing sufficient sensitivity and a flat frequency characteristic over most of the range was selected to simplify data processing. The band sound pressure level was determined by visually time-averaging the indications of a damped meter responding to the rms value of the microphone's output signal. The corresponding decay rates were determined from either time-interval measurements or the tracings of a high speed level recorder, a warble-tone signal being used for excitation of the reverberation room. Finally, the band sound power level was computed from the observed band sound pressure level and the decay rate, and these one-third-octave band sound power levels constitute the basic acoustic data presented in this report. (See Appendix III.)

The air flow conditions were monitored and adjusted with respect to mass flow rate and are reported in terms of standard cubic feet of air per minute. To accomplish this, high pressure air at about 90-100 psig was passed through a regulating valve and then a float-type flow meter. By adjusting the pressure at the flow meter in proportion to the air temperature observed there (effectively adjusting the air density), the flow meter became direct reading in SCFM. (See Appendix I.1 for details.) A second pressure regulator followed the flow meter to reduce the air pressure to the appropriate value for each nozzle under test. Thus, the pertinent flow data, which accompany the acoustic

data, are given in terms of standard cubic feet of air per minute (SCFM) and for most of the muffler experiments, flow values of 25, 50, and 100 SCFM were used. Table 1 provides an approximate guide to the mean jet velocities corresponding to the above mass flow values through the 0.500, 0.707, and 1.000 inch diameter smooth approach nozzles.

TABLE 1

APPROXIMATE JET VELOCITY FOR SMOOTH APPROACH NOZZLES

(Given in Mach number for c_0 = 1130 feet per second)

Mass Flow,	Nozzle Diameter, inch		
SCFM	0.500	0.707	1.000
100	1.00	0.50	0.25
50	0.50	0.25	0.12
25	0.25	0.12	0.06

The static pressure upstream of the nozzle was observed with a bourdon-tube pressure gauge. It was only intended to indicate the order of magnitude of the pressure ratio across the nozzle in the vicinity of critical pressure ratio. Also, it was used to indicate if a silencer arrangement significantly altered the pressure ratio. However, it was never intended for more exact purposes such as, for example, determination of nozzle coefficient.

The raw compressed air was not controlled with respect to temperature but rather used at whatever temperature the compressor, located out-of-doors, delivered. This arrangement resulted in temperatures ranging from 70° to 130°F observed at the flow meter. The actual air temperatures prevailing at the experimental nozzles were the thermodynamic result from the several expansions downstream of the flow meter. Generally, the final temperatures ranged from perhaps 20°F above to 50°F below room temperature.

Other researchers have investigated the relationship between jet temperature and total sound power. High temperature jets, all other parameters being equal, are definitely noisier. The magnitude of this temperature effect does not appear to have been satisfactorily explained on firm theoretical grounds. In any event, it is not very large for moderate changes in temperature. Roughly speaking, it seems proportional to some function (perhaps square root) of the ratio of the absolute temperature of the jet to the absolute temperature of the ambient air. Estimating from one reasonably successful empirical correction factor (C_4 of Reference 3), a change in temperature of over 100°F would be required to produce a 1.0 db change in sound power near room temperature ambient conditions. Consequently, the shifts in sound power due to the fortuitous temperature variations during the experiments reported here would be less than a decibel and therefore may be neglected. The experimental results confirm this presumption; sound power measurements on the same simple nozzle, conducted during cold winter

weather and warm spring weather, failed to disclose any consistent differences attributable to temperature.

During the planning phase of these experiments, an important consideration involved whether to design the air supply for blowdown operation or for continuous operation. Inasmuch as the reverberation room requires perhaps 15 seconds to reach equilibrium conditions and the audio-frequency spectrometer only permitted serial reading of the acoustic levels for the several frequency bands, it was decided that continuous operation for ten or fifteen minutes at a time was essential. Therefore, a compressed air system capable of supplying 100 SCFM continuously was obtained. Experience amply justifies this decision.

Another particularly pertinent consideration is the matter of scaling the results reported here to full-sized jets. Obviously, this research dealing only with model jets cannot provide a direct answer outside of the size range actually tested. Thus on this point, it is necessary to rely on the same types of arguments and comparisons presented, for example, by Sperry. He showed excellent correlation among the results for cold model jets as small as 0.533 inch in diameter and hot, conical nozzles four inches in diameter over a wide range of flow velocities. (Reference 3, Figure 14b, page 144.) Fair correlation was also shown for full-sized engines (Reference 3, Figures 17a, 17b, pages 150-151) expecially when taking into consideration the difficulties of collecting acoustical data in the field and the acoustical complexity of a complete, operational jet engine compared to a model nozzle.

This scaling has been demonstrated only for the case of simple converging nozzles, not for the variety of configurations which might be encountered among run-up silencers and not for many of the configurations described in the following sections of this report. However, the geometrical size of the jets has been taken into account by a strictly linear scaling, that is, mass flow per unit area, and total sound power per unit area. The temperature scaling, founded empirically, involved only physical parameters associated with the equation of state of the working fluid (γ, P, T) not geometrical parameters. Thus, tentatively, it seems reasonable to postulate that a similar separation of "variables" could be expected to apply to more complex configurations and that linear scaling with respect to geometry would represent a satisfactory first approximation.

Many of the experiments reported here have been conducted with nozzles of different size and at different mass flows. In this way, one can obtain some indication of the scaling relationships to be expected.

3.3. REFERENCE EXPERIMENTS

Some of the first measurements attempted dealt with ascertaining if the shape of the sound power spectrum and the relationships with respect to

nozzle diameter and mass flow, postulated above, actually could be realized experimentally in the reverberation room. Figure 5 is the experimentally obtained counterpart of Figure 4; it presents the measured sound power spectra with 100 SCFM air flow through smooth approach nozzles 0.500, 0.707, and 1.000 inch in diameter.[4] Obviously, the general situation predicted in Figure 4 has been found to hold experimentally. Consequently, one may proceed with the experiments dealing with silencing along the general lines discussed in subsections 3.1. and 3.2.. However, Figure 5 also reveals certain clearly discernable differences from Figure 4 which should be discussed, verified, and explored further.

Before continuing with a more detailed analysis of Figure 5, it is convenient to consider Figure 6 which presents the same experimental data for the 0.500 inch smooth-approach nozzle superimposed on a generalized spectrum for jet noise. (Also see Figure 2.) The generalized spectrum curve has been translated laterally to yield a best fit by eye in the vicinity of the peak at 6300 cps. The agreement is excellent. Previously, the total sound power level (assuming $K = 0.60 \times 10^{-4}$) was computed for the corresponding operating parameters using Equation (1) and found to be 115.5 db as indicated in the "broadband" column of Figure 6. Experimentally, the broadband sound power level up to 20 kc totals 114.4 db. However, it is evident that the experimental measurements have not been carried to high enough frequencies to include all of the contributions to the total sound power level. By assuming continued -2 db per octave behavior and including contributions so postulated through 480 kc, one obtains another 1.3 db or 115.7 db total experimental sound power level. The resulting difference of only 0.2 db between theory and experiment is much smaller than can be reasonably expected, considering the accuracy of the assumptions entered into the calculations. If the experimental peak frequency is taken as 6300 cps, the corresponding Strouhal number becomes 0.23, a value which agrees well with the literature. (See References 4 and 5.)

Perhaps an even better fit to the data could be obtained in Figure 6 by shifting the generalized curve slightly toward lower frequencies, however, the consequences would be small. Actually, the lower-frequency portion of the generalized curve was positioned more in accordance with the precept that when measuring continuous-spectrum sound, one is apt to error only by observing levels which are too large.

[4]Adjacent to each experimental curve are two types of reference information in the form of Roman numerals followed by Arabic numerals, II.10, III.37, which direct one to the appropriate figure or chart in Appendixes II and III. Appendix II contains illustrations of the nozzles and other configurations, such as screens or muffler shells, involved in the test. Appendix III presents the experimental sound power data in numerical form. When two or more Appendix III reference numbers appear with a single curve, it means that these several sets of data have been averaged (arithmetic average of decibels) to yield that particular curve.

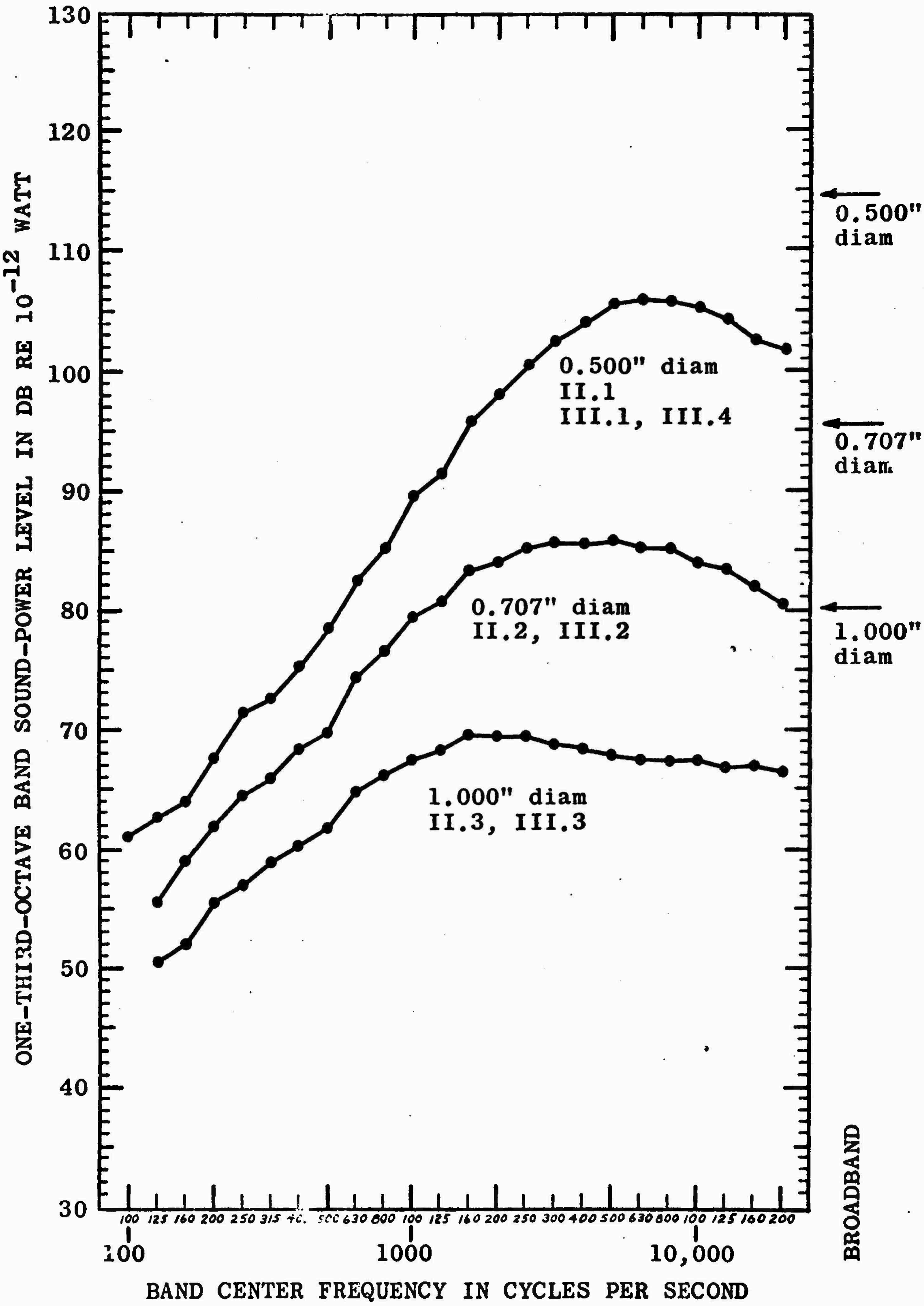

FIGURE 5. EXPERIMENTAL SOUND POWER SPECTRA FOR SIMPLE NOZZLES OPERATED AT 100 SCFM.

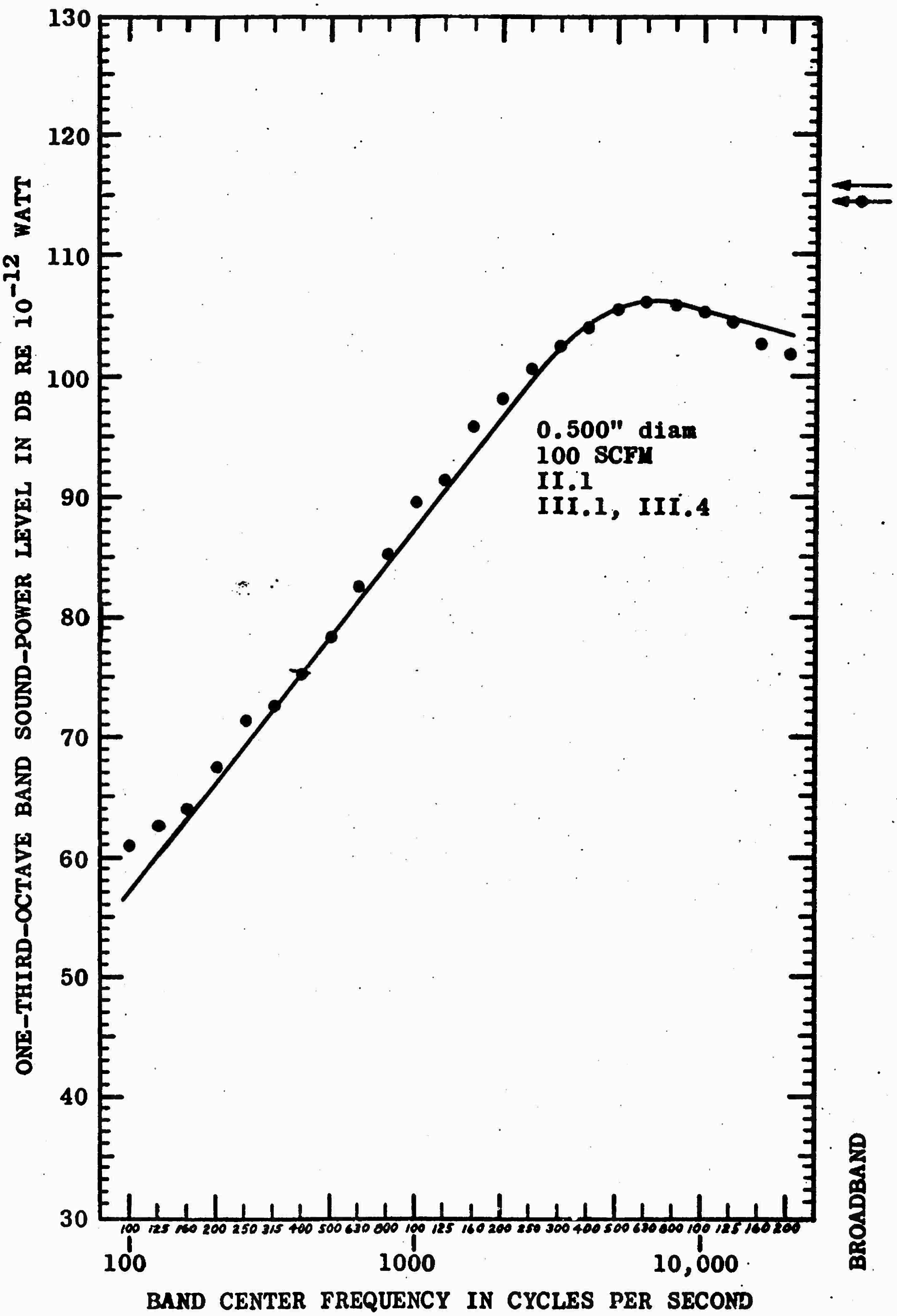

FIGURE 6. COMPARISON OF EXPERIMENTAL AND GENERALIZED SPECTRA.

Both the lowest and the highest two bands suggest divergence from the generalized curve. It is not clear at present whether this divergence is real or just an artifact. Both cases occur at the extreme ends of the frequency range available with the present measuring system and it is known that experimental accuracy decreases at both extremes because, among other reasons, the reverberation room gradually becomes less perfectly diffuse. At the low frequency end, the meter fluctuations become wilder and averaging them by eye may also contribute appreciably greater error than elsewhere in the spectrum. In addition, compressor noise and other interfering noises are more troublesome at the lowest frequencies. Furthermore, the signal levels at these lowest frequencies are 45 to 50 db below the peak levels which fact begins to strain the dynamic range available in the instrumentation. More elaborate precautions and procedures can provide somewhat improved accuracy of measurement at both extremes of the frequency range but higher accuracy seemed unnecessary for this particular research program. Generally, the absolute accuracy of the sound power measurements is considered to be about ±1 db while the relative accuracy is of the order of a few tenths of a decibel. Repeatability, especially for simple nozzles, is excellent as a comparison of III.1 and III.4 shows; the average difference between these two sets of data is about 1.0 db while the overall sound power levels only differ by 0.2 db. Many similar examples of nearly prefect repeats have occurred throughout the research but they will not be cited explicitly.

The accuracies discussed above contrast with the much lower accuracies ordinarily obtainable in field measurements and resulting from spacial-integration problems in free-field measurements. Reference 9 suggests that ±5 db is more representative of the absolute accuracy achieved in the determination of total radiated sound power level by the free-field method.

Returning to a detailed comparison of Figures 4 and 5, we recall that a stepwise reduction in total power level by 21 db was predicted in Figure 4. However, Figure 5 demonstrates a reduction of 18.9 db between the 0.500- and 0.707-inch nozzles, and 15.4 db between the 0.707- and 1.000-inch nozzles, omitting any adjustment for unmeasured high-frequency portions of the spectra. Evidently, the experimental reductions are somewhat smaller than predicted from simple theory and they become increasingly smaller as jet velocity decreases.

Figure 5 clearly indicates one reason for this result. The spectrum is less sharply peaked in the case of the 0.707-inch diameter nozzle than for the 0.500-inch diameter nozzle. This flattening trend is even more pronounced in the case of the 1.000-inch diameter nozzle. Moreover, various repeated measurements varify that these observed effects are real and not just some occasional happening.

At least two possible reasons for these divergences between simple theory and experiment are apparent.[5] Perhaps the situation described by simple theory must be limited to velocities approaching sonic velocities. However, Reference 4 states that the total power relationship generally is valid down to Mach 0.3 at least. A second distinct possibility is that the two larger nozzles in some manner fail to completely reproduce the jet turbulence situation implied by the simple theory. A partial answer to this latter possibility can be found by comparing the noise spectra for several different mass flows through the same nozzle; a test condition which will be presented shortly.

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[5] After the experimental work had been completed and a considerable portion of this report written, Dr. Alan Powell (University of California, Los Angeles, California) suggested during a discussion that the observed departures from simple jet noise were due to flow separation occurring at the sharp internal corner of the calming chamber upstream of the model nozzles. He gave an incomplete reference by memory to some early British research which was said to have demonstrated that if upstream flow separation were scrupulously avoided then one would obtain precisely the noise predicted for simple jets all the way from sonic velocity down to as low a velocity as one choses to investigate. Subsequent discussion with Dr. Richard Waterhouse (The American University, Washington, D. C.), coauthor of Reference 9, revealed that he also was not familiar with this information asserted by Dr. Powell. The geometry of the calming chamber reported in Reference 9 included a sharp corner upstream which might introduce a question of flow separation there also.

The author has not yet located the material referred to by Dr. Powell in order to judge for himself its pertinence to the rest of the discussion in Section 3.3 of this report. Our experimental apparatus had been stored by the time of the discussion reported above and so it was not possible to directly investigate the matter although it would not be difficult to check this matter in future research.

Be that as it may, the possibility of unintentional flow separation upstream of the model nozzles does not affect significantly the main body of this research dealing with silencing. The silencing effectiveness of any particular configuration was always compared to the noise generated by an unsilenced nozzle operating under the same flow conditions. Furthermore, if flow separation and the associated excess noise is as hard to avoid even in the laboratory as Dr. Powell suggests, then one would expect that many practical engineering flow situations must typically include some excess noise due to flow separation. Thus practical silencers would have to cope with this excess noise. Moreover, there remains a very difficult area of research in learning to control such a noise mechanism and to quantitatively predict the characteristics of the noise associated with flow conditions involving partially-separated flow.

The three smooth approach nozzles used to collect the data in Figure 5 (see II.1, II.2, and II.3) were intended to be a matched set in order to provide a direct comparison with Figure 4. Their areas are exactly in the ratios of 1:2:4. They are all polished smooth-approach nozzles having constant approach radii.[6] It was expected that these three nozzles, as fabricated, would each produce, (1) an essentially rectangular velocity profile, (2) little or no vena contracta, and, (3) small-scale turbulence at the nozzle exit. These expected conditions have not been confirmed by experiment but are rather confidently predicted, at least to the approximation needed in this research, from general knowledge about fluid flow.

There has been some mention in the literature (Reference 7, for example) that the detailed characteristics of a nozzle's boundary layer, as it develops upstream of the exit, may have a significant effect on the resulting noise spectrum. With this possibility in mind, the approach radii for the set of three smooth-approach nozzles have been placed in inverse ratio to the exit area. The idea was to cause a particle of fluid following the boundary streamline (inviscid flow) to experience more nearly the same angular acceleration for all three nozzles when they are operated at the same mass flow. Whether or not this selection of radii was a good idea is still open to question.

Figure 7 presents the spectra obtained with the 0.500-inch diameter smooth-approach nozzle for a range of mass-flow values. Since the same nozzle was used for all of these spectra, any question about the similarity among nozzles has been eliminated. Of course, the boundary layer is not deliberately manipulated but remains whatever is the natural consequence of the particular flow conditions used. As before, the gross spectral shifts from one condition to the next are approximately those expected but as mass flow (and velocity) decreases, the spectrum shape progressively flattens until the 25 SCFM spectrum has become an almost straight line sloping +3.5 db per octave. Certainly, this upward sloping spectrum must reach a maximum and finally decrease above some higher frequency which lies outside the range of our results. This behavior was not anticipated, and evidently, the simple theory does not strictly apply to the jet noise situations encountered in our experiments at the lower flow velocities. Both Figure 5 and Figure 7 support this contention.

For more detailed comparisons, Figure 8 shows the predicted spectra for the same conditions as were used to obtain the experimental results presented in Figure 7. The broadband power levels agree fairly well down to 40 SCFM, a result which perhaps can be observed more readily from Table 2. However, significant departures in spectral shape became evident at 60 SCFM. In Figure 7, it is obvious again that measurements have not been carried to

[6] Slightly higher nozzle efficiencies would be expected if elliptic approach configurations had been used but this was not considered worth the additional fabrication complexity.

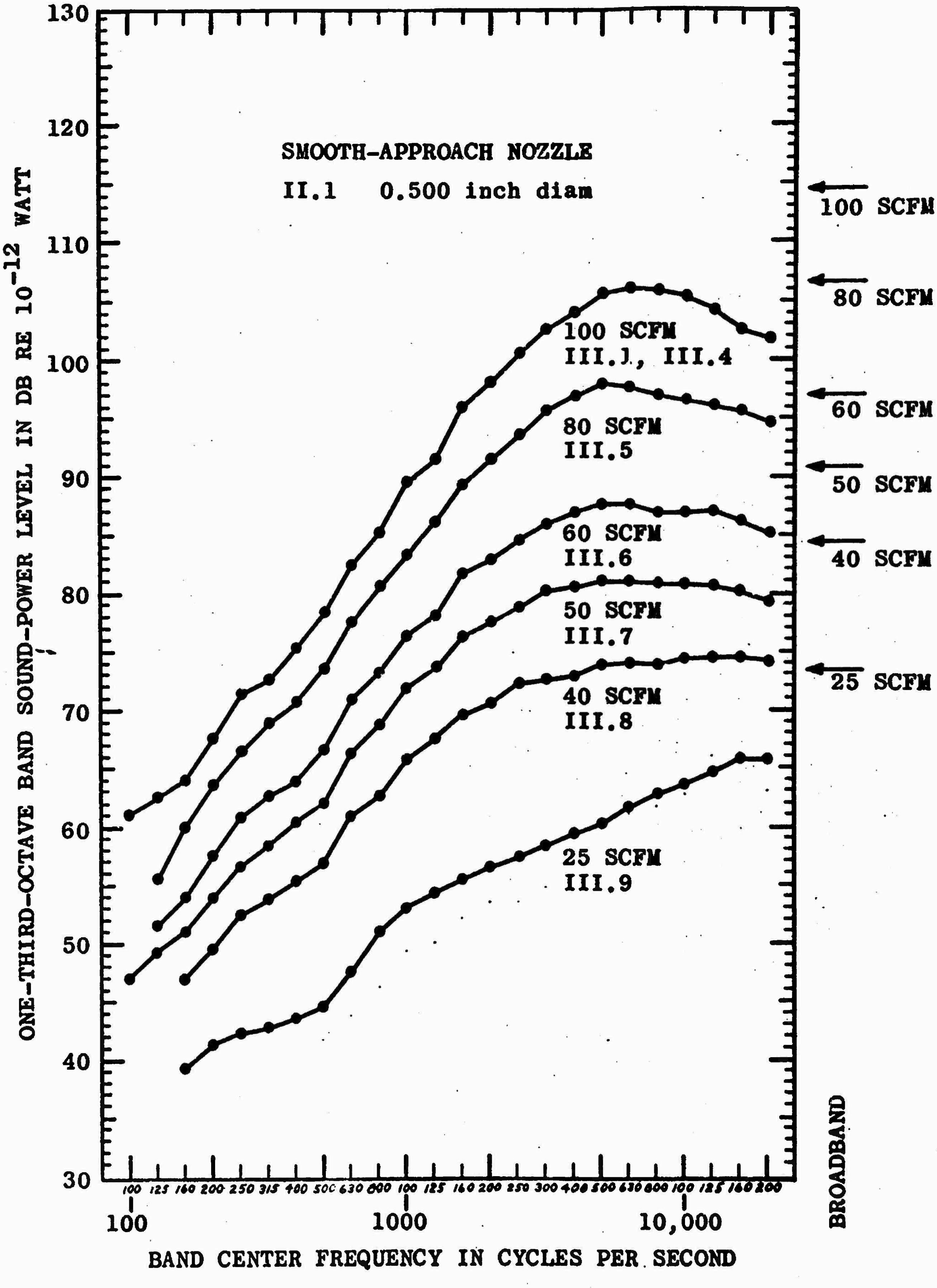

FIGURE 7. EXPERIMENTAL SPECTRA FOR NOZZLE AT VARIOUS FLOWS.

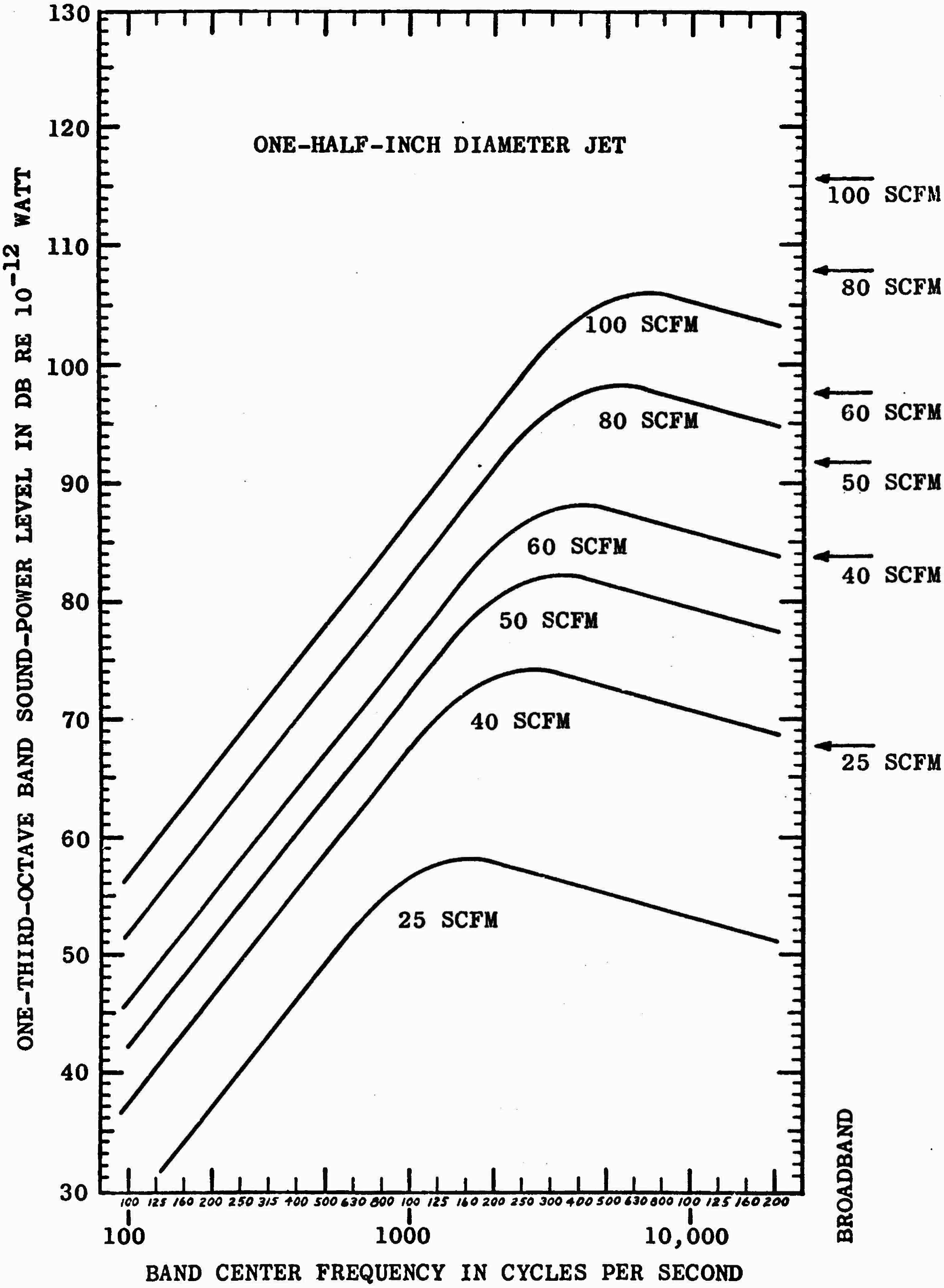

FIGURE 8. EXPECTED SOUND POWER SPECTRA AT VARIOUS FLOWS.

high enough frequencies to permit computation of the total sound power levels. However, this unobserved portion of the spectrum could only

TABLE 2

TOTAL SOUND POWER LEVELS FOR 0.500-INCH NOZZLE AT VARIOUS FLOWS

Total Sound Power Level	Mass Flow Rate, SCFM					
db re 10^{-12} watts	100	80	60	50	40	25
Predicted	115.5	107.7	97.7	91.5	83.7	67.5
Observed*	114.4	106.5	97.0	90.8	84.3	72.7

*Broadband level, not total sound power level.

increase the disparity in total power noted above. Figure 9 demonstrates this aspect of jet noise in a somewhat different manner. By plotting total or broadband sound power level against mass flow on a logarithmic scale, strict conformity with the eighth power law would produce a straight line of appropriate slope. At high mass flows, the eighth power behavior is confirmed but at lower mass flows, the velocity dependence grows progressively smaller. Impending departure from a straight line of U^{+8} slope has become evident at 60 SCFM. The slope decreases to no more than U^{+6} at the lower flow values and, in reality, may be even less.

One might expect that the acoustic behavior at low flow velocities would approach that for ventilation system grills. A search of the pertinent literature failed to uncover sound power data which could be used for direct comparison with the present research. However, the acoustic data presented in Reference 10 imply that U^{+4} to U^{+6} might be appropriate in a velocity range from Mach 0.01 to 0.04. Moreover, both Reference 10 and Reference 11 suggest a flat to upward sloping spectrum across the entire audiofrequency range for low flow velocities, certainly not the broadly peaked spectrum shape of Figure 2.

A more complete investigation of total sound power level and spectrum shape from essentially zero flow velocity up to sonic flow velocity would constitute a legitimate subject for future research. With respect to this research, the non-conformity with simple theory need not frustrate the research because a considerable amount of silencing (downward from sonic velocity) can be explored before the departures cause any difficulty. However, the departures do pose problems of a more subtle nature, somewhat as follows:

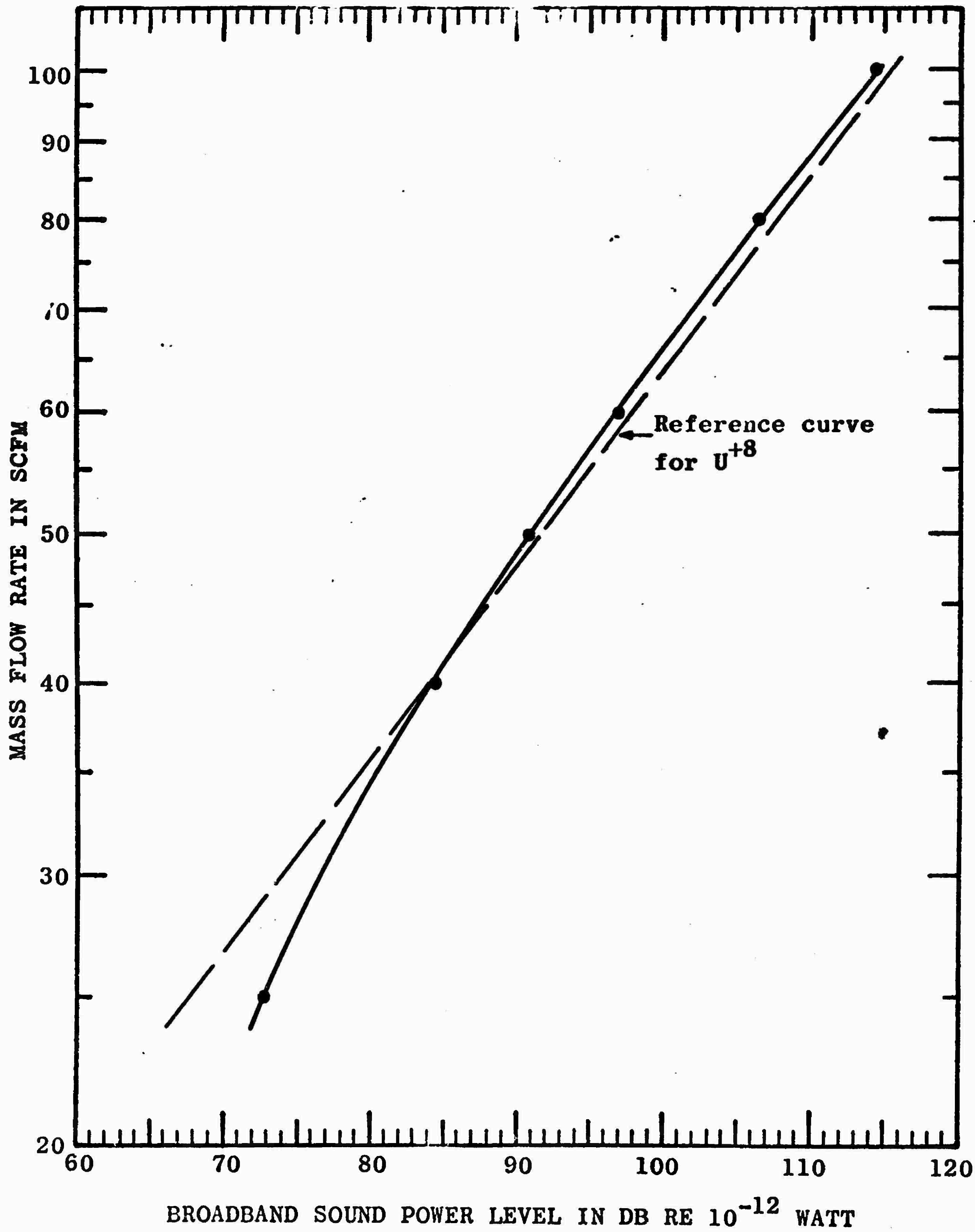

FIGURE 9. BROADBAND POWER LEVEL AS A FUNCTION OF MASS FLOW RATE.

(1) The maximum amount of silencing obtainable remains somewhat undetermined. It can be large but not as large as one would predict from Equation (2); and

(2) The changes in spectrum shape may prevent a calculation of total sound power in some cases. This situation complicates comparisons among the many test conditions and generates lingering doubts about higher-frequency behavior which will go unobserved in these studies but which, due to scaling, may still be important in full-scale designs.

Nevertheless, these model studies can still accomplish most of their purpose.

Another point of initial concern is how well our reverberation room results agree with small-nozzle results obtained by other investigators using the free-field method for determining total sound power. It was decided to explore this question by duplicating Sperry's smallest nozzle (Reference 3, nozzle 100; our nozzle II.4) and testing it throughout an overlapping range of flow conditions. This nozzle differed from our smooth-approach nozzles by the presence of a short, straight cylindrical section. The spectral data are given in III.11 - III.14 but are not graphed since the distributions are similar to those in Figure 7.

In order to continue the comparison, Figure 10A duplicates Sperry's presentation for the appropriate mass flow range (Reference 3, Figure 6, page 94) and compares our results (indicated by crosses) with Sperry's results (indicated by open circles). The agreement is seen to be excellent; even the minor deviations from a "theoretical" curve assumed by Sperry are duplicated. Figure 10B shows a similar comparison where the crosses now represent results obtained from our smooth-approach nozzles. Figure 10 clearly demonstrates agreement of broadband radiated acoustic power measured by the free-field and reverberant-field methods. It also demonstrates that the geometrical differences between our smooth-approach nozzles II.1 - II.3 and Sperry's nozzle, II.4, are completely inconsequential with respect to broadband sound power, at least in this experiment.

The remarks above deliberately used the term "broadband" instead of "total" relating to the sound power quantities. It has been pointed out earlier that the measurements reported here do not extend above 20,000 cps. Thus we are prevented from taking higher-frequency acoustic power into account unless from some independent information, we may assume its nature.[7] In the case of the reverberation-room measurements reported here, the high-frequency limitation results from the rapidly increasing absorption con-

[7]In the case of simple jets operated near sonic velocity, we may hypothesize on the basis of Figure 2. However, in general, and more specifically for most of the experimental situations treated in the following sections of this report, there exists no such information to serve as a guide.

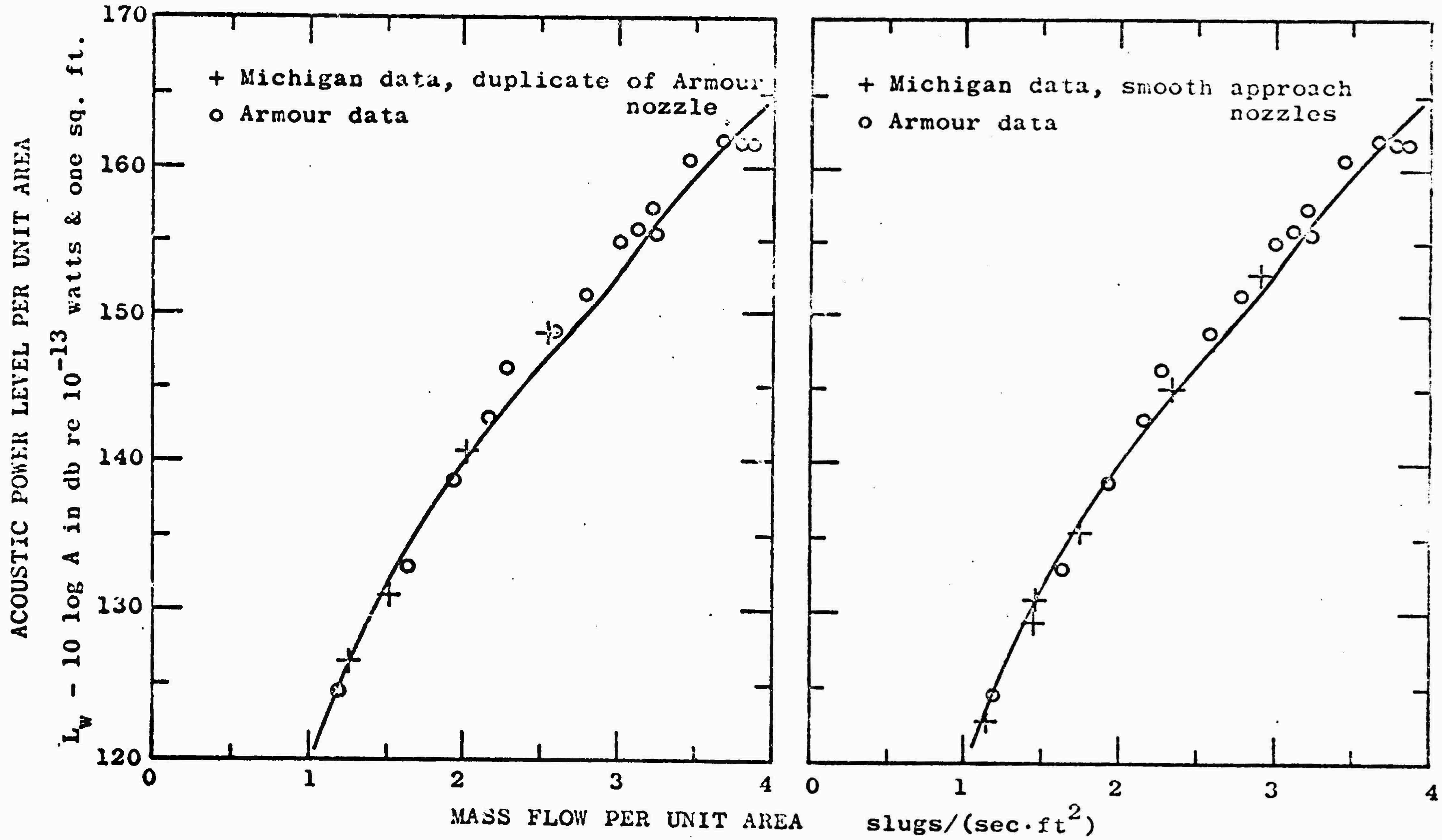

Figure 10A

Figure 10B

FIGURE 10. COMPARISON OF ACOUSTIC POWER MEASURED IN A FREE FIELD AND A REVERBERANT FIELD.

tributed by water vapor in the atmosphere. Some extension of range might be achieved by using a smaller reverberation room for the highest-frequency measurements and by operating with dry nitrogen, for example, instead of air.

The free-field measurements, either out-of-doors or in anechoic rooms are high-frequency limited also in one way or another. The absorption due to water vapor is important in these cases also, and unless measured in situ and simultaneously, will contribute much uncertainty to the results. Recently, precision values of absorption due to water vapor have been published for a frequency range from 2000 cps to 12,500 cps (Reference 12) but at higher frequencies, reliable values do not seem to be available. Out-of-door precision measurements are strongly influenced by wind and thermal gradients and absorption along the ground surface. Anechoic room measurements are often confounded by uncertain wall reflection characteristics at very high frequencies. In addition, the instrumentation system, often the microphone itself, contributes a high-frequency limitation. The microphones on many sound level meters are only sensitive to 10,000 cps. Even if the microphone possesses high-frequency sensitivity, its range of flat response is limited by its physical dimensions.[8] Above some limiting frequency, band measurements and correction band by band become necessary, a practice not often followed. There is indication that significant high-frequency contributions do occur out to 100,000 cps in the near field of operational jet aircraft. (See Reference 13.)

It may be validly argued that in the practical far-field situation the extra attenuation due to water vapor in the intervening atmosphere is always present to some extent, and it is beneficial in reducing the noise, so why concern ourselves with it? However, from a research viewpoint, where we inquire about the nature of the source and may wish to scale the results to other sizes, such an argument is inadmissible.

Another concern in the present research is the lowest acoustic levels which can be measured accurately in the reverberation room. That is, various interfering noises will be generated by the compressor and air-regulation system, other activity around the building, and by the movement of the reflector vane within the reverberation room. It seemed impossible to achieve a directly comparable operating condition with an air flow but without jet

[8] In Reference 3, it appears that the total pressure levels were taken directly using a MR-104 microphone which presumably has characteristics similar to the W.E. 640-AA condenser microphone. From physical size, one would expect a significant decrease in sensitivity by about 10,000 cps and so the extent to which the high-frequency sound is taken into account in the broadband measurement depends in detail upon the individual spectrum shape and the microphone's response curve. Suffice it to state here that, whatever the limitations, they are of small and similar magnitude for the comparisons drawn in Figure 10.

noise. One attempt was simply to remove the nozzle insert and to operate the system with a 3.750-inch opening into the settling chamber. Figure 11 (also III.15 and III.16) presents the results, where it can be seen that the potentially interfering noise occurs mainly in the low-frequency portion of the spectrum.

Comparison with some of the earlier spectra, for example, Figure 7, shows that the levels reported by Figure 11 in the lowest bands are higher than when a nozzle is present. This result is probably due to the larger area through which noise within the settling chamber can propagate into the reverberation room when a nozzle is not present. It is probably not justifiable to assume a strict area ratio effect either. In any event, Figure 11 represents the maximum background level that can be produced, and normally this level will be lower.

In some experimental situations, the sound power level, starting with the lowest bands, decreases for several bands and then begins to rise. If this occurs at low sound-power levels, we generally assume that the measurement is indicating background noise and that the true jet noise spectrum is represented only after the band levels begin to rise again. The data in Appendix III report the measured values regardless of level, but when drawing spectral graphs, these downward sloping results are omitted. Figure 7 and Appendix III.9 demonstrate this handling of the data.

Background level experiments similar to the one described above have been conducted many times during the course of the experiments and have yielded very similar results. There is no point in adding to the volume of this report by including all of such data. Experiments have been conducted also in which the positions of the several screens within the settling chamber have been shifted or even completely removed. No significant alterations of the spectra for nozzle II.1 at either 100 SCFM or 50 SCFM have been observed as a result of such changes in the settling chamber. The detailed data for these experiments have been eliminated from this report also. The implication from all of these tests, however, is that the noise generated by various nozzles is essentially independent of the detailed constructional arrangement of the upstream portions of the apparatus and of the detailed settings of the several control valves. Only the nozzle geometry or the downstream configuration seems to affect the noise; this is the desired result.

One final observation about repeatability of results belongs in this section. During the course of the research, nozzle II.1 was insufficiently secured. It blew out of its seat and suffered rather severe damage near the O-ring groove on the upstream side. As a precaution, a new II.1 nozzle was machined and measured. No significant differences in acoustic performance could be found between the original nozzle, the damaged nozzle and the new nozzle.

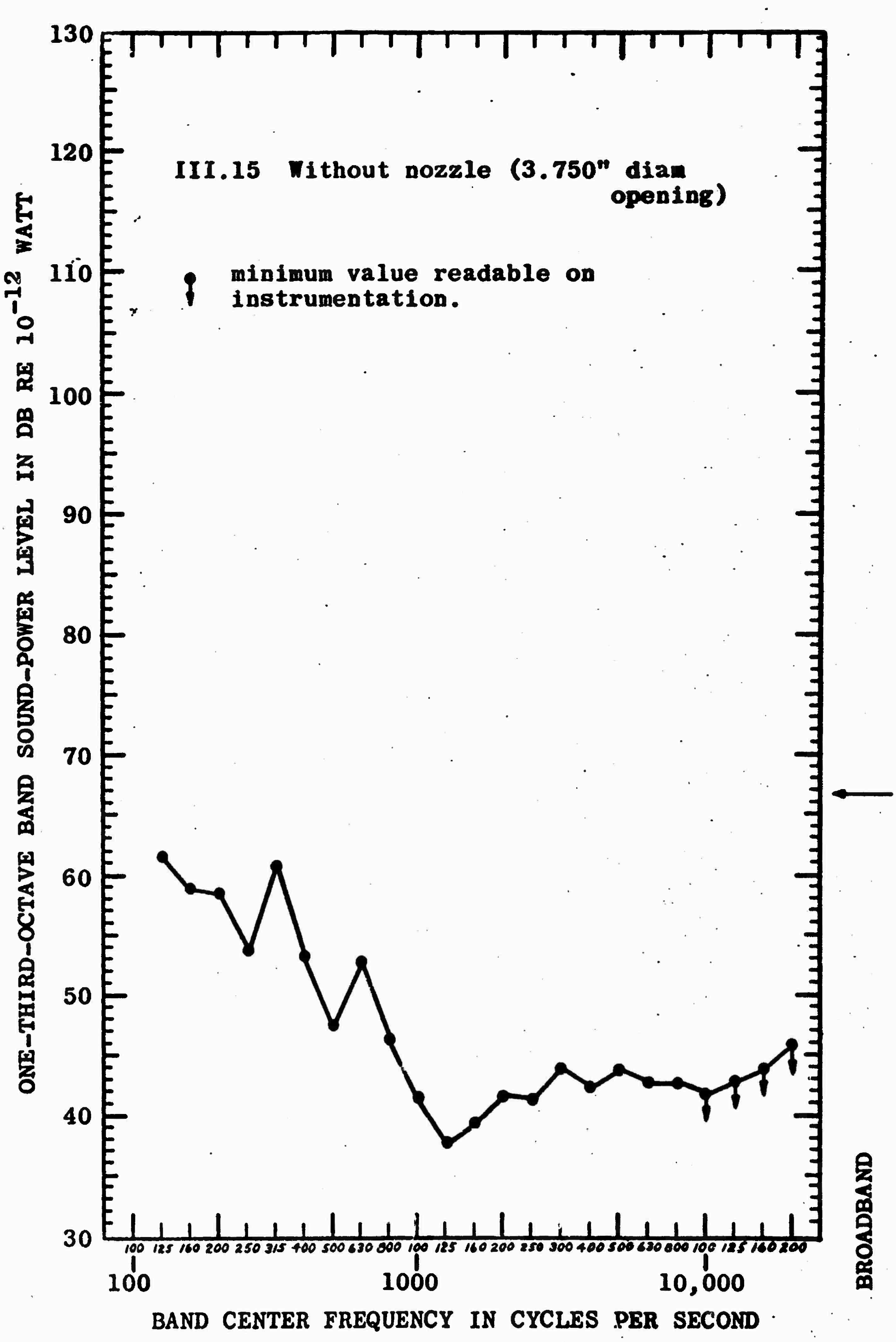

FIGURE 11. BACKGROUND NOISE WITHOUT NOZZLE AT 100 SCFM.

A slightly lower broadband sound-power level (111.8 extreme minimum value vs. 114.4 db) was observed during the coldest winter weather with nozzle II.1 at 100 SCFM. The entire spectrum shifted downward slightly but maintained its shape. The observed upstream pressure also dropped slightly, 13.9 psig vs. 14.4 psig. The reverse trend occurred as the weather became warmer in the spring. This "seasonal" shift was very gradual; measurements taken a few days apart always agreed within a few tenths of a decibel and were considered to be within experimental error. The maximum amount of shift was small compared to the silencing results sought but larger than one would expect from known shifts in the jet temperature. Of course, the moisture content of the compressed air changed with season as well as the moisture content in the reverberation room. The correction for absorption in the reverberation room was noticeably larger in winter than in summer. In any event, the exact cause of the slow variation in broadband level was not ascertained, but probably can be explained by combined moisture and temperature effects.

SECTION 4

EXPLORATION OF CHANGES IN NOZZLE CONFIGURATION

In order to proceed with research directed toward ground silencing, it was desirable to explore the effects on the radiated sound power of changes in nozzle configuration. Some of the experiments, such as those incorporating a short tubular extension, relate directly to silencing because a similar arrangement may be necessary to connect a jet engine to a muffler. Other experiments, for example, those with sharp edge nozzles, are more indirectly related; they contribute insight into how the flow may be manipulated with acoustic objectives rather than fluid-engineering objectives in mind. Configurations resembling the "daisy-petal" types of flight silencers (see, for example, Reference 14) were intentionally omitted as being too marginally related to the ground silencing problem.[9]

4.1. SHARP EDGE NOZZLES

Several experiments were conducted with sharp edge nozzles (II.5, 6, and 7) in contrast to the smooth approach nozzles (II.1, 2, and 3). Fluid-dynamic engineering seems devoted to streamline design in order to minimize pressure losses. Most reported acoustical investigations relating to flow-generated noise appear to have been influenced by this emphasis on streamlining and have utilized flow geometries which are representative of streamlined design. It therefore seemed appropriate in our research to produce some "bad" flow designs and to examine the acoustical consequences.

Figure 12 compares the acoustical spectrum of a 0.500-inch diameter sharp edge nozzle with that of the same diameter smooth-approach nozzle, both operated at 100 SCFM. The sharp edge nozzle is just detectably noisier across the entire spectrum with a broadband level of 116.2 db compared with 114.4 db for the smooth-approach nozzle. The shapes of the spectral distributions are very similar. As expected, the upstream pressure required to maintain the mass flow rate is considerably higher for the sharp edge nozzle than for the smooth-approach nozzle; 20.4 psig compared to 14.4 psig. The absence of any pronounced shift in the location of the spectral peak suggests that the sharp edge nozzle has little or no vena contracta for approximately sonic flow.

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[9] Technically, however, model studies of flight silencer configurations by the reverberation room method would seem worthwhile. Determination of scaling factors using already published full-scale results, and further development of such silencers based on the more controlled laboratory experiments, both appear to be suitable topics for future research.

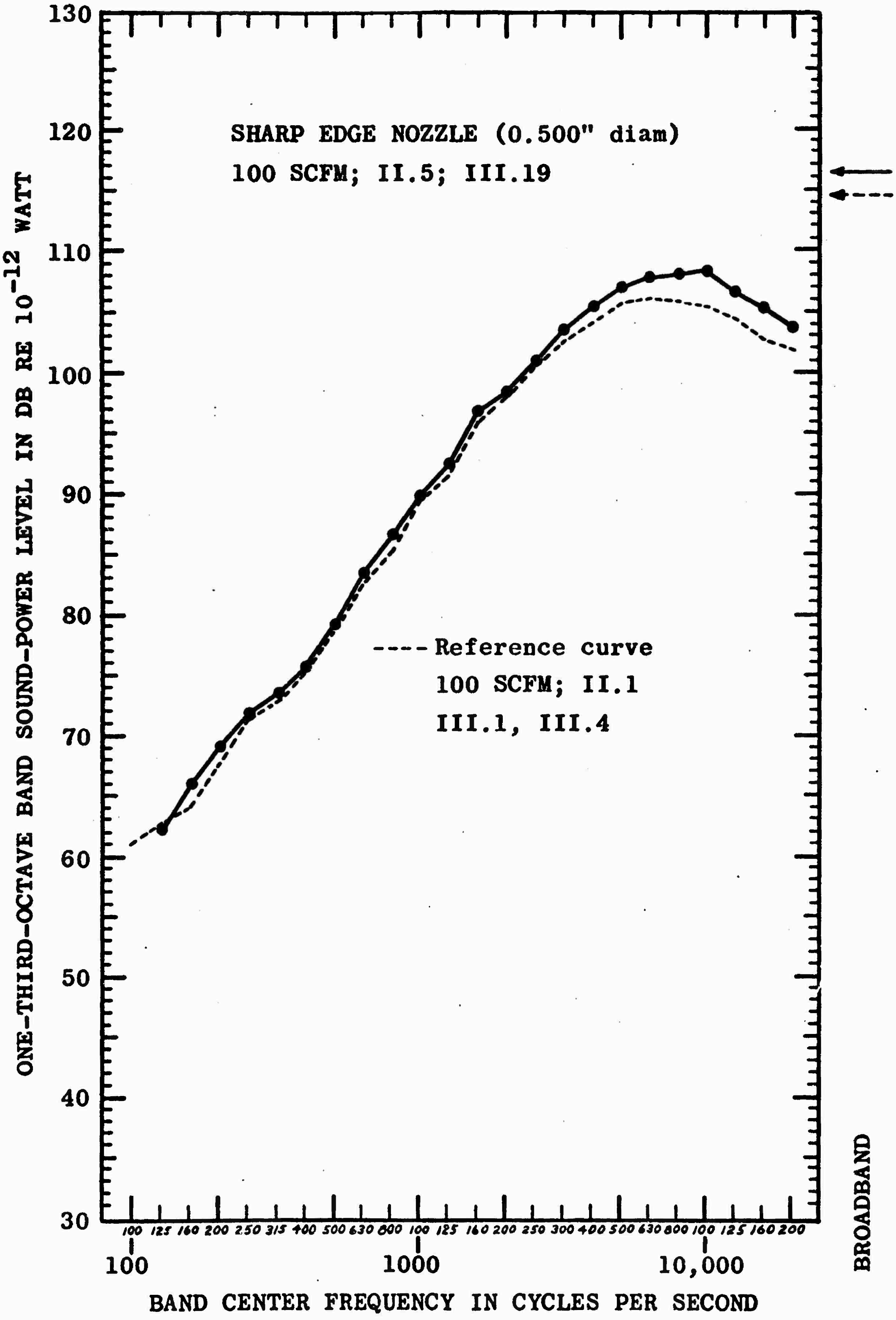

FIGURE 12. SPECTRUM OF SHARP EDGE NOZZLE NEAR SONIC VELOCITY.

However, Figure 5 has indicated that spectral-peak location may be a less sensitive measure of jet dimension than simple theory would suggest.

When the flow velocity through a sharp edge nozzle is made subsonic, the acoustical picture changes drastically as Figure 13 demonstrates. Now the sharp edge nozzle is clearly noisier than its smooth-approach counterpart. Of more significance is the evidence of tonal generation indicating the activation of an additional acoustic source mechanism. The evidence of tonal generation is the disproportionately large sound power levels observed in the one-third octave bands centered at 10 kc and 20 kc. The broadband sound power has been increased almost three orders of magnitude in this case by changing from a "streamlined" approach geometry to a "non-streamlined" approach geometry.

Analysis by one-third octave bands does not provide sufficient resolution to distinguish among a very narrow band of frequencies, a tone with slightly unsteady frequency, and a pure tone of constant frequency.[10] The presence of two high levels spaced one octave apart suggests a first and second harmonic. The frequency range does not extend far enough to encompass higher harmonics if they exist. Also, it is not possible to decide from these simple observations whether the assumed second harmonic is due directly to the wave form of the additional source mechanism or due to the amplitudes at the nozzle being large enough to engender finite amplitude phenomena. However, the fact remains that this sharp edge nozzle, operated at subsonic flow, has introduced an additional source mechanism into our considerations regardless of its detailed nature.

The portion of the spectral curve lying between 630 cps and 6300 cps (Figure 13) is reminiscent of a simple jet noise spectrum but for a somewhat higher velocity flow than represented by the smooth-approach reference curve. If we make the reasonable assumptions that the peak would occur at 6300 cps, that the flow velocity times the area of the jet remains constant, and that $S_m = 0.23$, then application of Equation (3) predicts a jet diameter of about 0.4 inch and a velocity of about Mach 0.8 for this 0.500-inch sharp edge nozzle at 50 SCFM. These computed values are sensible. They are consistent with the interpretation that the vena contracta has resulted in a minimum jet diameter 20% smaller than the physical dimension of the nozzle and, of course, the velocity increased proportionately.

The appearance of tonal phenomena at subsonic but not sonic flow velocities suggests a positive feedback mechanism whereby fluctuations occurring downstream propagate upstream, influence the flow approaching the nozzle, and hence reinforce the fluctuation. The characteristic period involved in the tonal generation is not readily predictable from the data available inasmuch as the velocity of sonic propagation in the jet relative to a fixed frame of reference is different in the upstream and downstream directions.

[10] Much narrower-band analyses are feasible technically but were not considered appropriate to the present research program.

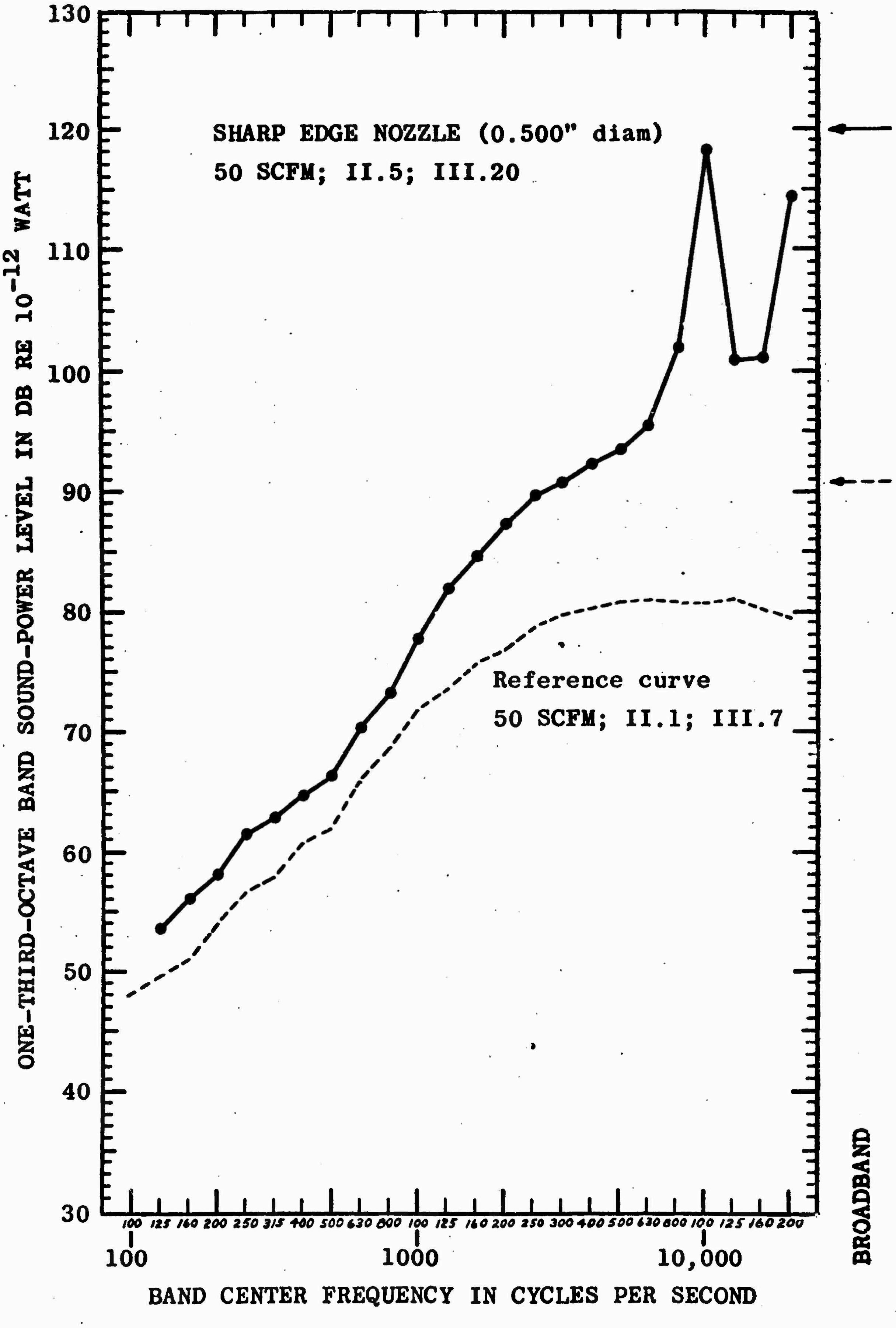

FIGURE 13. SHARP EDGE NOZZLE WITH SUBSONIC FLOW.

Figure 14 displays the spectra for a larger sharp edge nozzle, 0.707 inch in diameter, and the spectra for the comparable smooth-approach nozzle. In this case, the flow velocity is subsonic for both flow conditions and clear evidence of tonal generation occurs. Moreover, rather peculiar maxima have appeared almost two octaves below the predominant peaks. Also, the reference curve for 50 SCFM (0.707 inch diameter smooth-approach nozzle) demonstrates a sudden increase in the highest two frequency bands which suggests that tonal generation has not been completely avoided even with the smooth-approach designs. However, this was one of only two test situations with smooth-approach nozzles in which any hint of tonal generation appeared; see Figure 22 for the other case.

Figure 15 presents the corresponding acoustical information about a 1.000 inch diameter sharp edge nozzle. Tonal effects and other irregularities are still evident but are not nearly as pronounced as in Figure 14. Nozzles II.5, 6, and 7 were constructed to the same thickness, 0.375 inch, to restrict the thickness parameter to one arbitrarily selected value.

It would seem from these limited studies with sharp edge nozzles that large but still distinctly subsonic jet velocities are required to generate the strongest tonal phenomena. The frequency of the predominant peak shifts downward as flow velocity decreases. More elaborate analyses have not been attempted since the actual flow velocities pertaining to these experiments are not known precisely nor easily estimated. A more detailed investigation would be required to display clearly the relationships among the frequencies and amplitudes of the tones and the nozzle and flow parameters. Such a detailed investigation did not seem appropriate here. Nevertheless, even these limited data, demonstrating strongly enhanced sound power generation by certain sharp-edge subsonic nozzles, clearly warn us to beware of incorporating similar flow conditions into a muffler design. Enhanced sound generation of the magnitudes demonstrated could easily overwhelm other significant noise-reducing design features.

To investigate the role of the sharp edge and thickness of these nozzles in contributing the increased noise and tonal characteristics, several experiments were conducted using chamfered nozzle plates possessing the geometry shown in Figure II.8. The sharp right-angled corner upstream of nozzles II.5, 6, and 7 has been replaced by a 45° chamfer, also with sharp corners, and the width of the straight section has been reduced to only 1/32 inch. The acoustical results for two diameters of this chamfered nozzle configuration are displayed in Figures 16 and 17 along with the corresponding smooth-approach nozzle reference curves. In the case of a 0.500-inch diameter chamfered plate operated at 100 SCFM (see Figure 16), the sound power appears to have been slightly reduced compared to the corresponding smooth-approach nozzle. In a similar comparison, the sharp-edge nozzle (nozzle II.5; see Figure 12) produced a slight increase. These changes in the spectra are small but consistent enough to presume them to be real physical effects. The true cause of the reduction displayed in Figure 16 is unknown but it might be related to the detailed characteristics of the boundary layer.

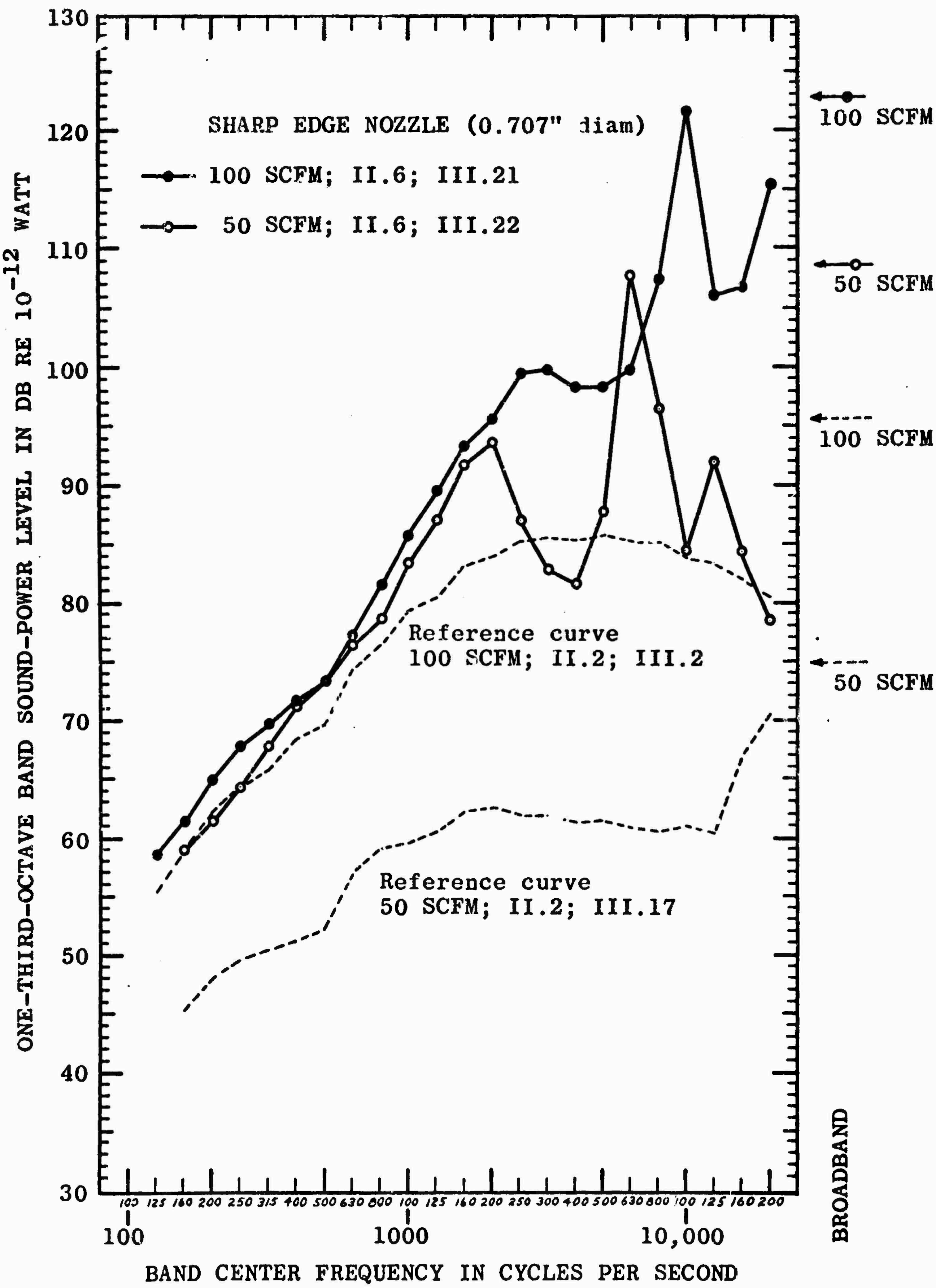

FIGURE 14. SHARP EDGE NOZZLE 0.707 INCH IN DIAMETER.

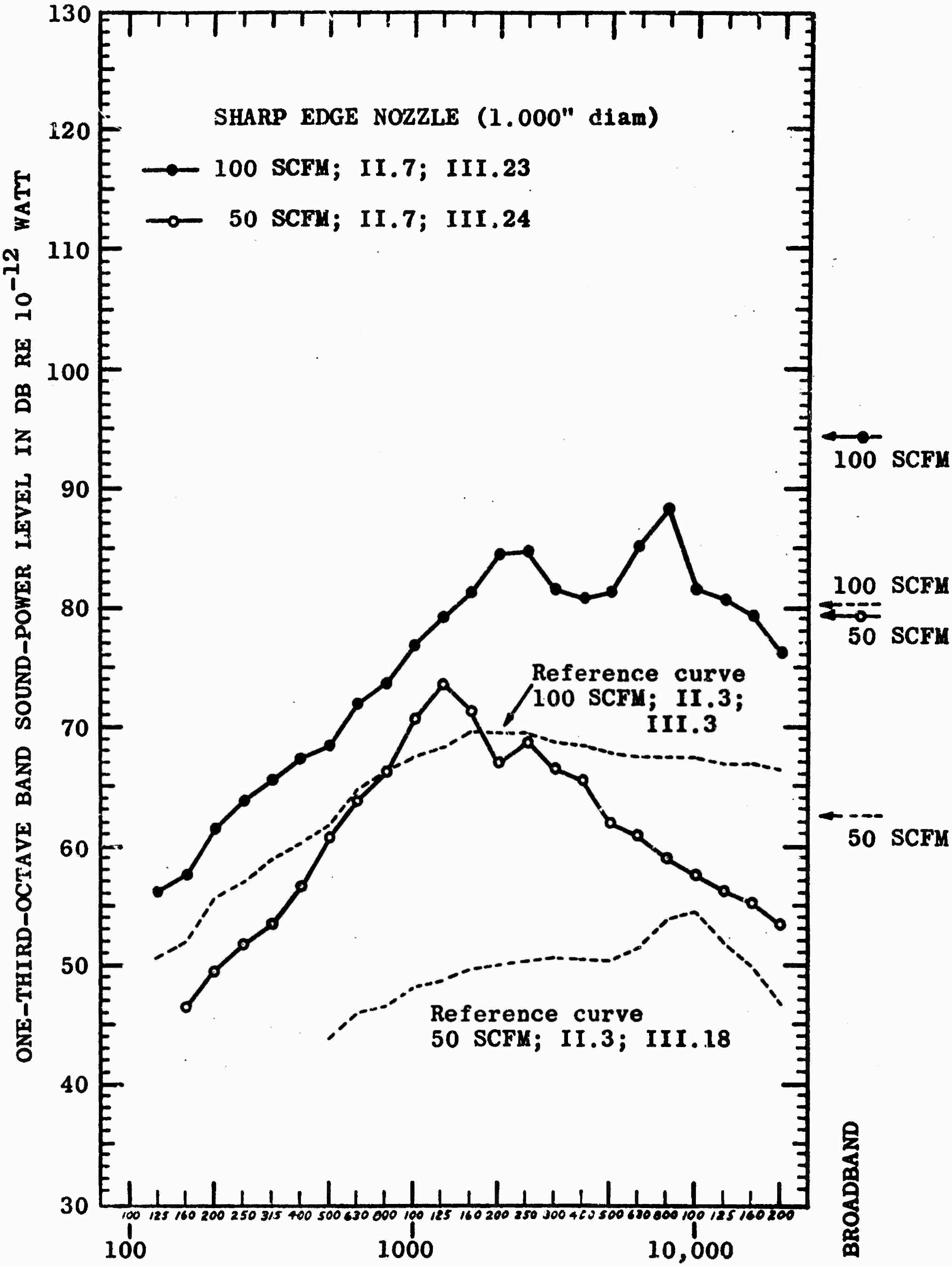

FIGURE 15. SHARP EDGE NOZZLE 1.000 INCH IN DIAMETER.

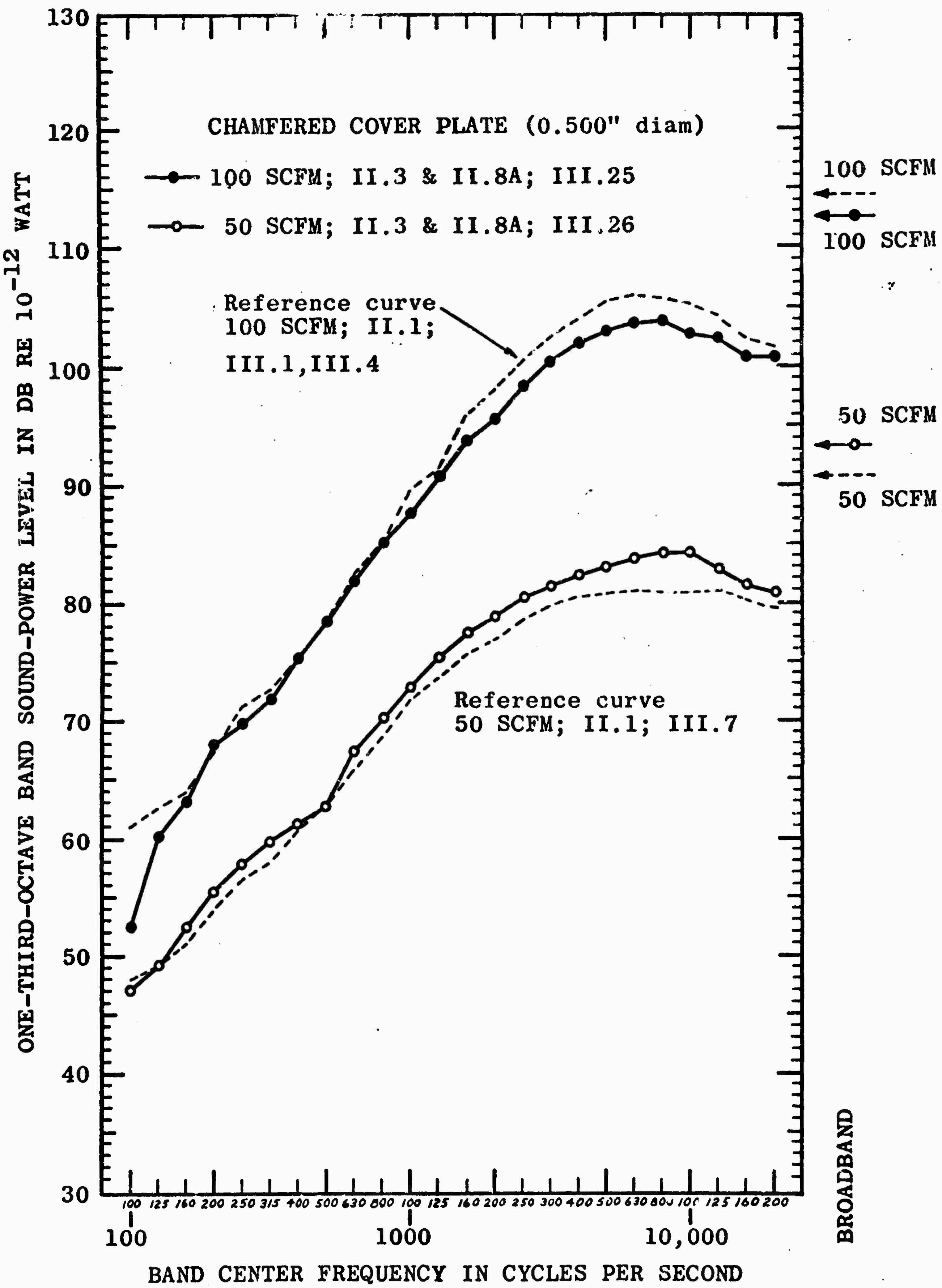

FIGURE 16. CHAMFERED COVER PLATE 0.500 INCH IN DIAMETER.

At lower flow velocity, the effect is reversed; the chamfered plate generates slightly more noise than a smooth-approach nozzle of the same diameter. No hint of tonal generation is evident in Figure 16 in contrast to the behavior of nozzle II.5 displayed in Figure 12.

Figure 17, for a 0.707-inch diameter chamfered cover plate, displays additional low velocity results. Both spectra for the chamfered plate lie somewhat above the reference spectra for the corresponding smooth-approach nozzle and they display no evidence of tonal generation.

Considered also, was the acoustical effect of reversing the chamfered cover plate so that the right-angled corner faced upstream while the chamfer faced downstream. This configuration was tested using the 0.500-inch diameter cover plate (nozzle configuration II.8A except with plate reversed) and the spectral data are given in III.56 and III.57. The results are insignificantly larger than when the chamfer faced upstream (see III.25 and III.26 for comparison); a new graph to supplement Figure 16 is unnecessary.

Taken as a whole, the results demonstrating increased levels for chamfered plates operating at subsonic flows are qualitatively consistent with the interpretation that the vena contracta results in an equivalent simple jet, slightly smaller in diameter and possessing a higher velocity than expected from nozzle dimensions alone. Whether or not such an explanation is quantitatively complete has not been determined. Furthermore, the tonal generation displayed by nozzles II.5, 6, and 7 with subsonic flow seems to be related in some way to the finite thickness of these nozzles. Finally, at or near sonic velocity, the broadband sound power level and the spectral distribution of the sound power appear to be almost independent of nozzle geometry; at least, within the scope of geometries reported here.

4.2. DIFFUSER

One of the first considerations with respect to ground runup silencing was the possibility of expanding the jet flow in a diffuser without introducing additional noise. (See the discussion pertaining to Figure 3.) Hopefully, the residual noise might be only the simple jet noise appropriate to the flow conditions at the outlet of the diffuser. Intuitively, such an elementary postulate of the acoustical behavior of a diffuser must be erroneous or successful mufflers of this type would have been constructed already.

The lack of reference literature on the acoustical behavior of diffusers argued for a direct test of the matter. To this end, the diffuser shown in Figure II.9 was fabricated. It has a smooth-approach geometry identical to nozzle II.1, a 0.500-inch diameter throat, and a 1.000-inch diameter exit. The total included angle is 14 degrees, chosen because it approximates the expansion angle of a free jet and it is close to the angles used in the diffuser sections of subsonic wind tunnels.

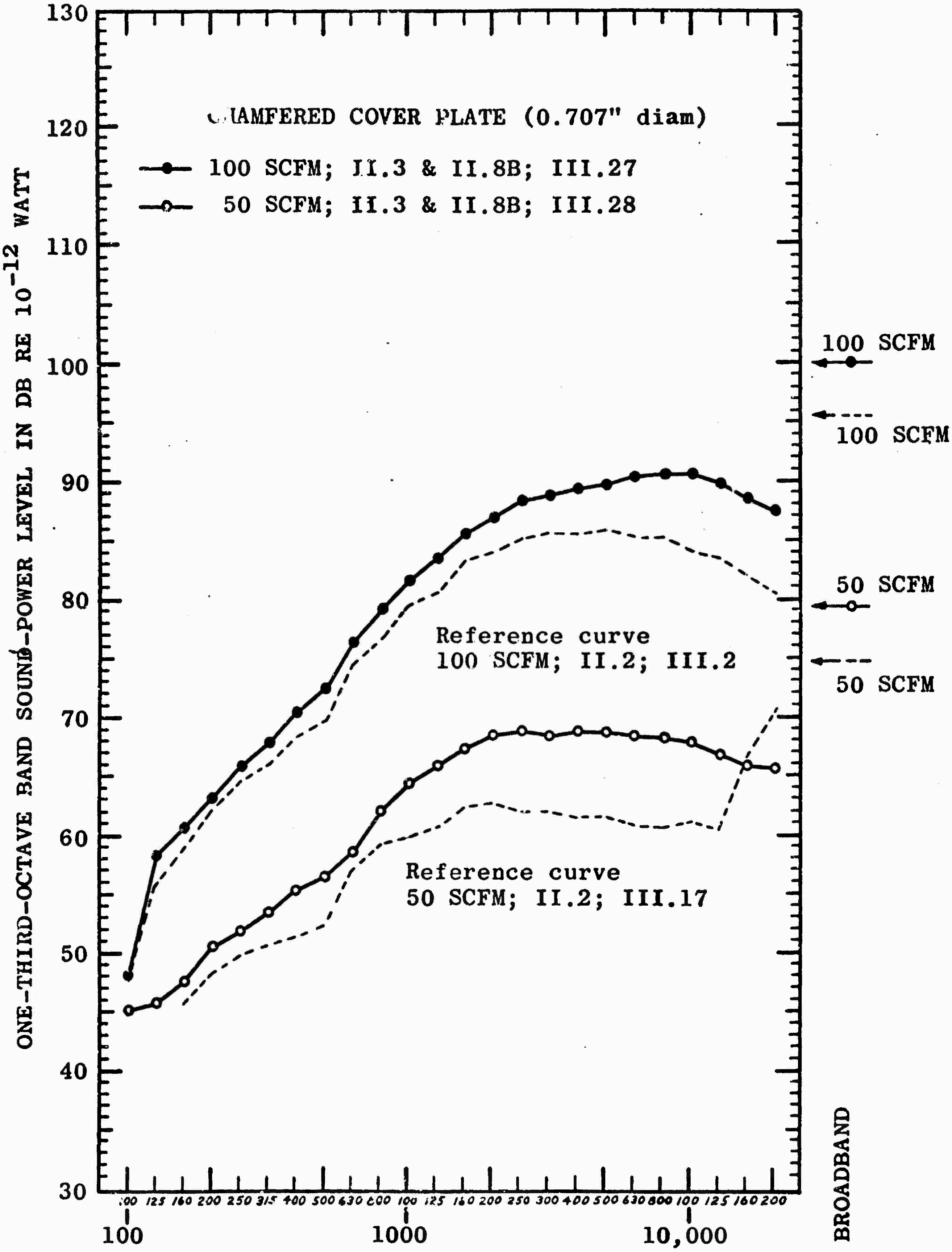

FIGURE 17. CHAMFERED COVER PLATE 0.707 INCH IN DIAMETER.

If this diffuser functioned according to the elementary postulate mentioned above, one would expect to start with the highest curve of Figure 4 and arrive at the lowest curve of Figure 4, that is, to experience a broad-band reduction in sound power level of 42 db. Figure 18 displays the actual acoustical behavior of the diffuser at 100 SCFM. Instead of decreasing, the entire spectrum shifts upward by nearly an order of magnitude and some evidence of tonal generation appears. In Figure 18, the appropriate experimental reference curves for smooth-approach nozzles have been plotted instead of the theoretical curves from Figure 4.

New mechanisms of noise generation have been introduced or the efficiency of noise generation by a simple jet has been increased or both. The sound power output is roughly 20,000 times the amount one might anticipate from the dimensions of the diffuser exit. Also, this amplification factor of 20,000 might be considered as representing the magnitude of the problem undertaken when trying to silence a jet.

Figure 19 displays the alterations of the spectra for the diffuser as the mass flow, and hence the flow velocity, is varied. As the mass flow rate is decreased a very pronounced tonal generation predominates the spectrum. This behavior is in marked contrast to that displayed in Figure 7 for a simple smooth-approach nozzle. Also note the apparent frequency stability of the tonal generation. It is suggestive of a mechanism which depends upon some parameter not varied in these tests such as a physical dimension of the diffuser; however, this remark is speculative until more evidence is available. If shock waves are involved, as suggested below, then the presence of harmonics probably is to be expected as a result of finite amplitude phenomena. Obviously, a simple diffuser of the type reported here is not useful as a silencer.

The acoustical results displayed in Figure 19 can be explained after a fashion. If one recalls the one-dimensional isentropic flow in a converging-diverging nozzle as discussed in any modern treatise on fluid flow (see Reference 15, Chapter 9 for example) and takes into consideration the pressure ratios appropriate to the mass flow rates used, then it is evident that a shock wave will occur somewhere within the diffuser. The axial location of the shock wave will shift as the mass flow rate is varied but shock-free flow will occur only for mass flow rates much smaller than any tested. Conversely, it appears to be impossible to design a simple diffuser which can start with critical pressure ratio and expand the flow in a shock-free manner to ambient pressure.

4.3. MULTIPLE NOZZLES

This group of experiments is concerned with how the sound power generated by two physically separated nozzles combines to yield the total radiated power. If each nozzle generates noise as a completely independent source, then the radiated power will be additive and the spectrum will merely in-

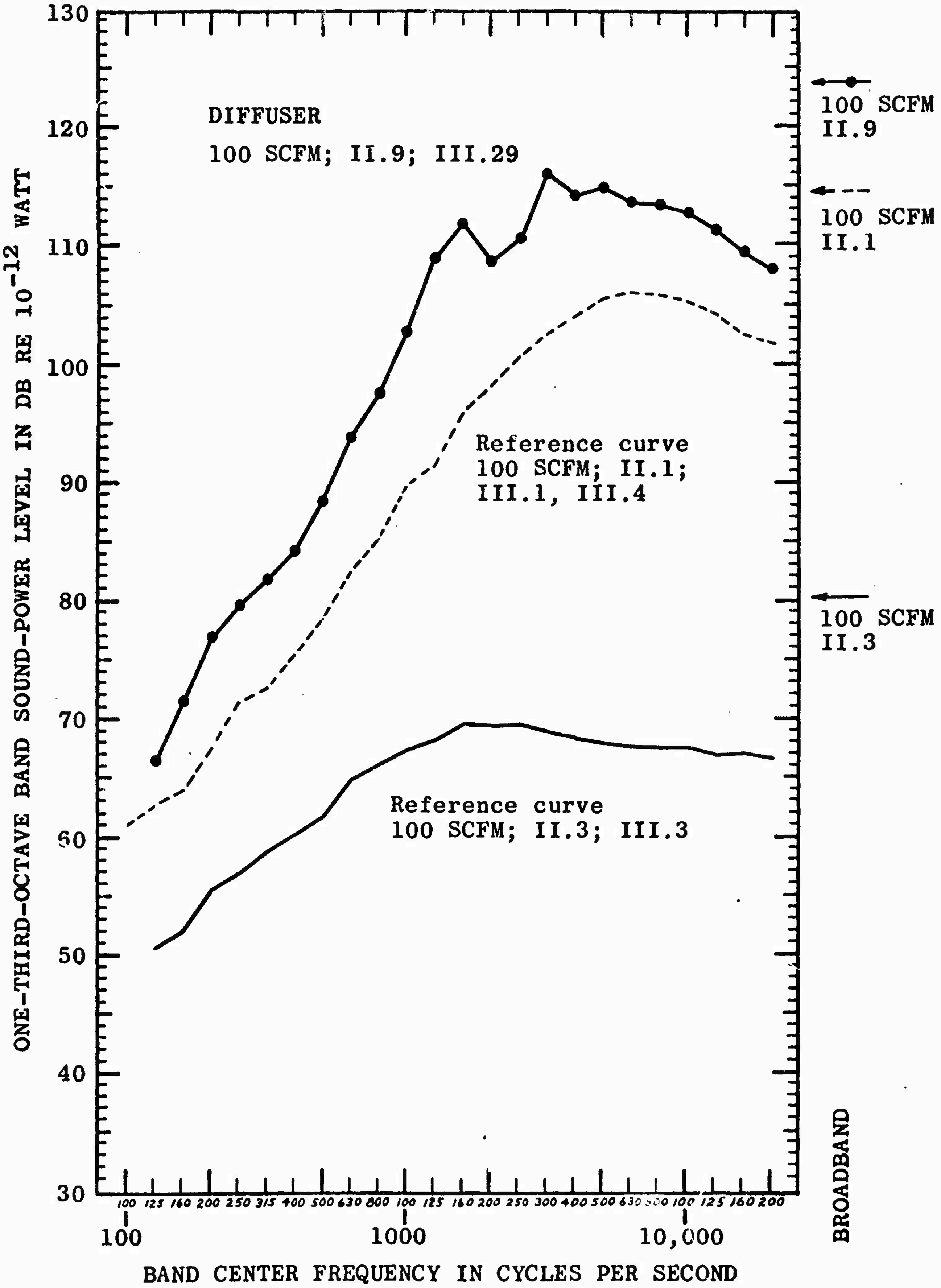

FIGURE 18. DIFFUSER FROM 0.500 INCH TO 1.000 INCH IN DIAMETER.

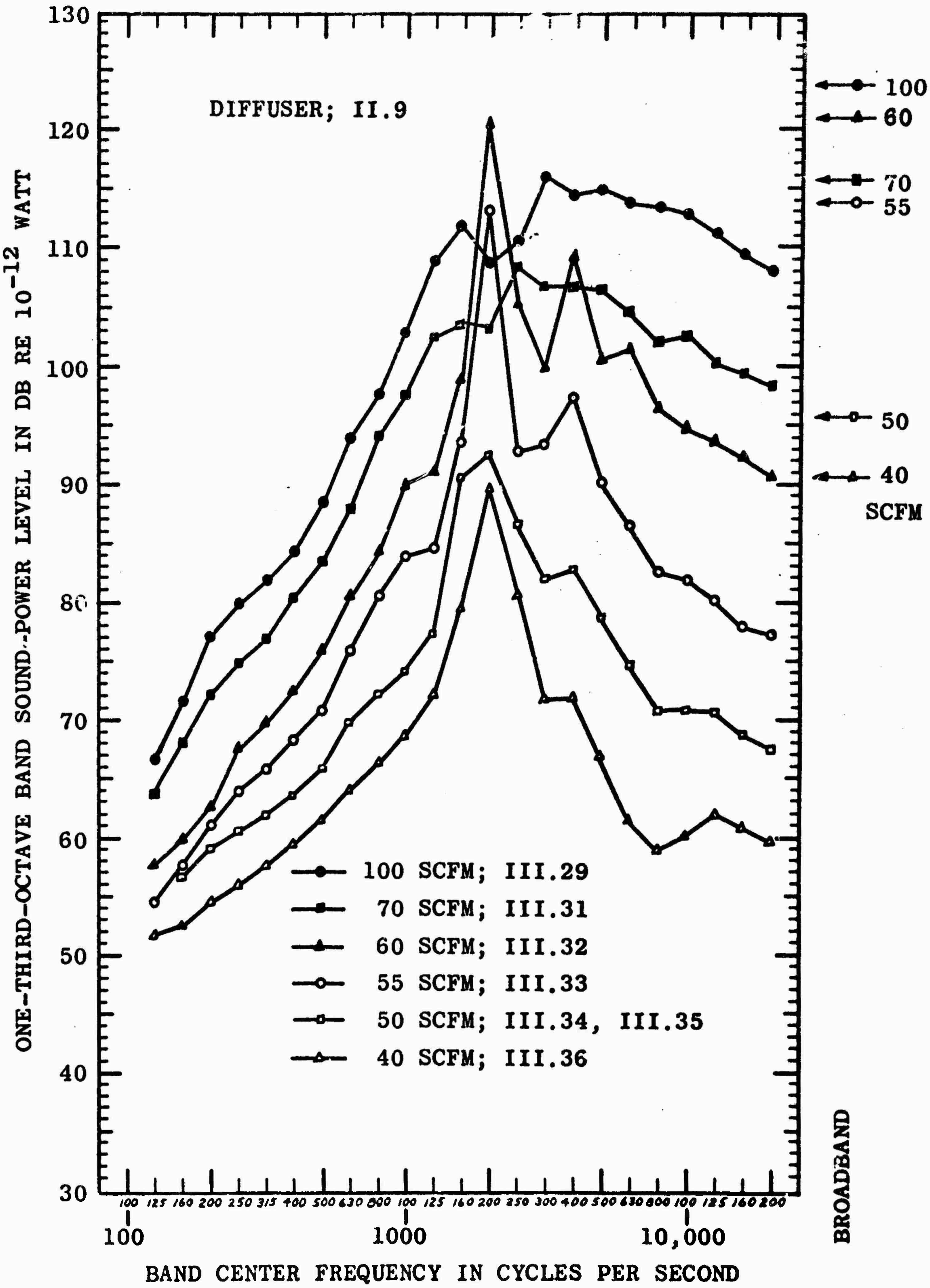

FIGURE 19. SPECTRA FOR DIFFUSER AT VARIOUS MASS FLOW RATES.

crease in level by 3 db. No other effect upon the spectrum would be expected to occur. On the other hand, if there were to be interactions between the two jets, somewhat different effects should be evident in the resulting spectrum. In one sense, these experiments bear directly upon the feasibility of using a sound cancellation phenomenon to achieve silencing. If multiple jets act differently from independent sources, a possibility would exist for minimizing the radiated sound by appropriate adjustment of phasing among the contributing jets.

If however, multiple jets act as completely independent noise sources, Equation (1) predicts the generation of the same total amount of sound power regardless of how many individual jets are used to accommodate the same total flow. It is presumed that the jet velocity is maintained constant and the multiple jets present the same total area as the equivalent single jet. In such a situation, Equation (3) predicts that the location of the spectral peak might be placed at any desired frequency higher than that for the equivalent single jet. Such a situation represents true frequency changing without any direct alteration in total radiated power. Jet silencing accomplished in this manner would necessarily depend upon some factor not included in the simple description of jet noise; for example, increased absorption of the air at high frequency, or less apparent loudness at some different frequency.

As a partial experimental investigation of considerations discussed above, the double smooth-approach nozzle illustrated in Figure II.10 was tested. The 2.500-inch separation of the nozzles was the largest distance readily achieved with our flow facility. It was considered large enough so that the flow from each nozzle would have expanded and decelerated markedly before the jets started to coalesce. Figure 20 presents the acoustical results. As indicated, the corresponding reference curves have been shifted upward by 3 db to test additivity of the radiated sound. The upper set of data points, for a flow rate of 100 SCFM, follows its reference curve rather closely. There are small divergences but they appear to be of doubtful significance. At 50 SCFM, the divergences are larger. The double nozzle appears to produce a flatter spectrum. The lower frequencies are somewhat enhanced while above 5000 cps a slight decrease is observed. Such behavior might be ascribed to the effective sources being separated by more than half a wavelength at high frequencies and less than half a wavelength at low frequencies; partial shielding occurring at high frequencies and cooperative interaction occurring at low frequencies. However, the evidence is not strong enough to justify much interpretation and the 50 SCFM flow condition falls in an acoustical region of comparatively low accuracy of measurement.

In order to observe the acoustical results of permitting the two jets to coalesce before they had expanded very much, nozzle II.11 was fabricated with only a one-inch separation. Figure 21 displays the corresponding spectra. The results are quite similar to those for the wider spacing presented in

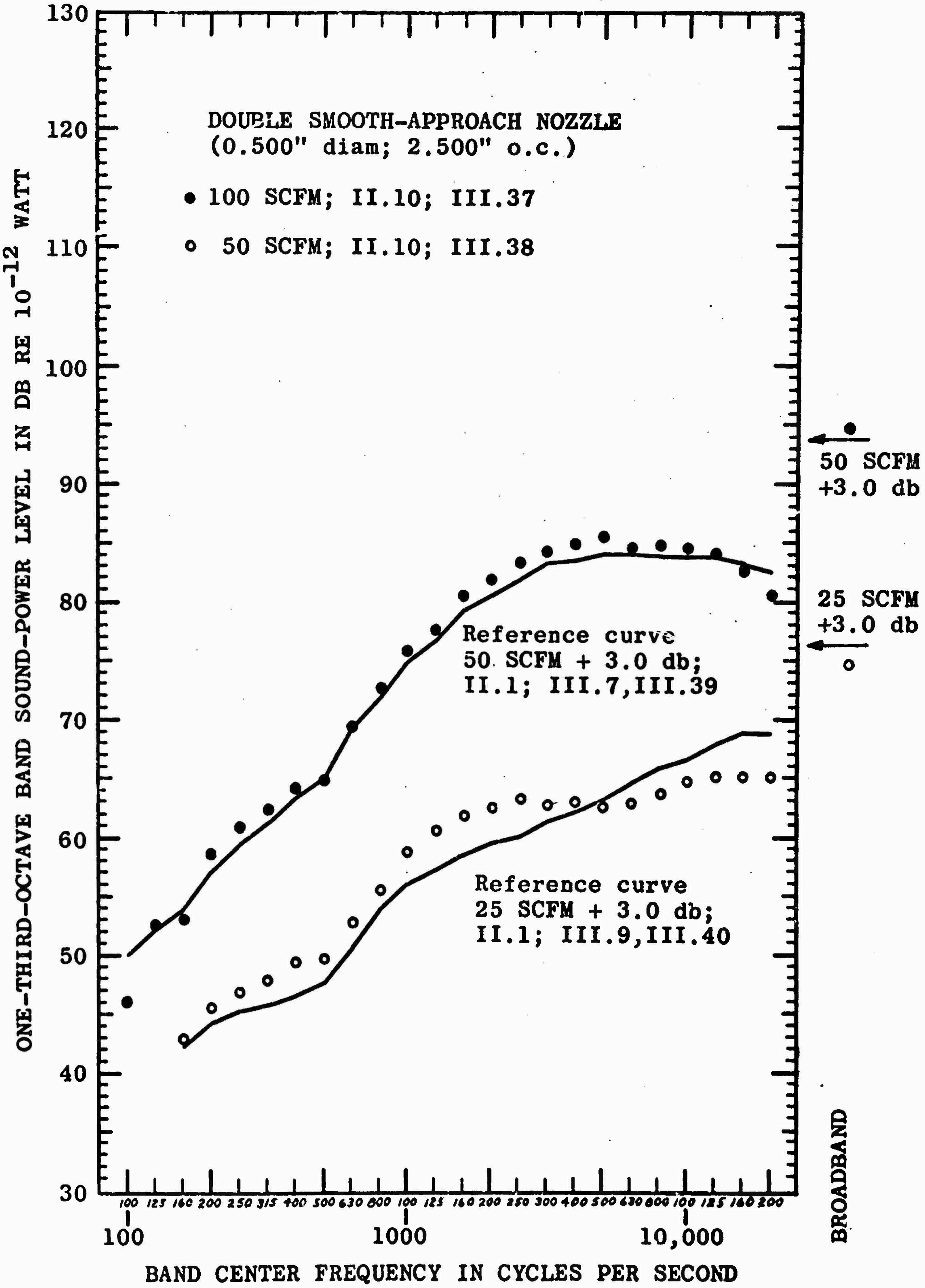

FIGURE 20. TWO 0.500 INCH DIAM NOZZLES 2.500 INCHES ON CENTERS.

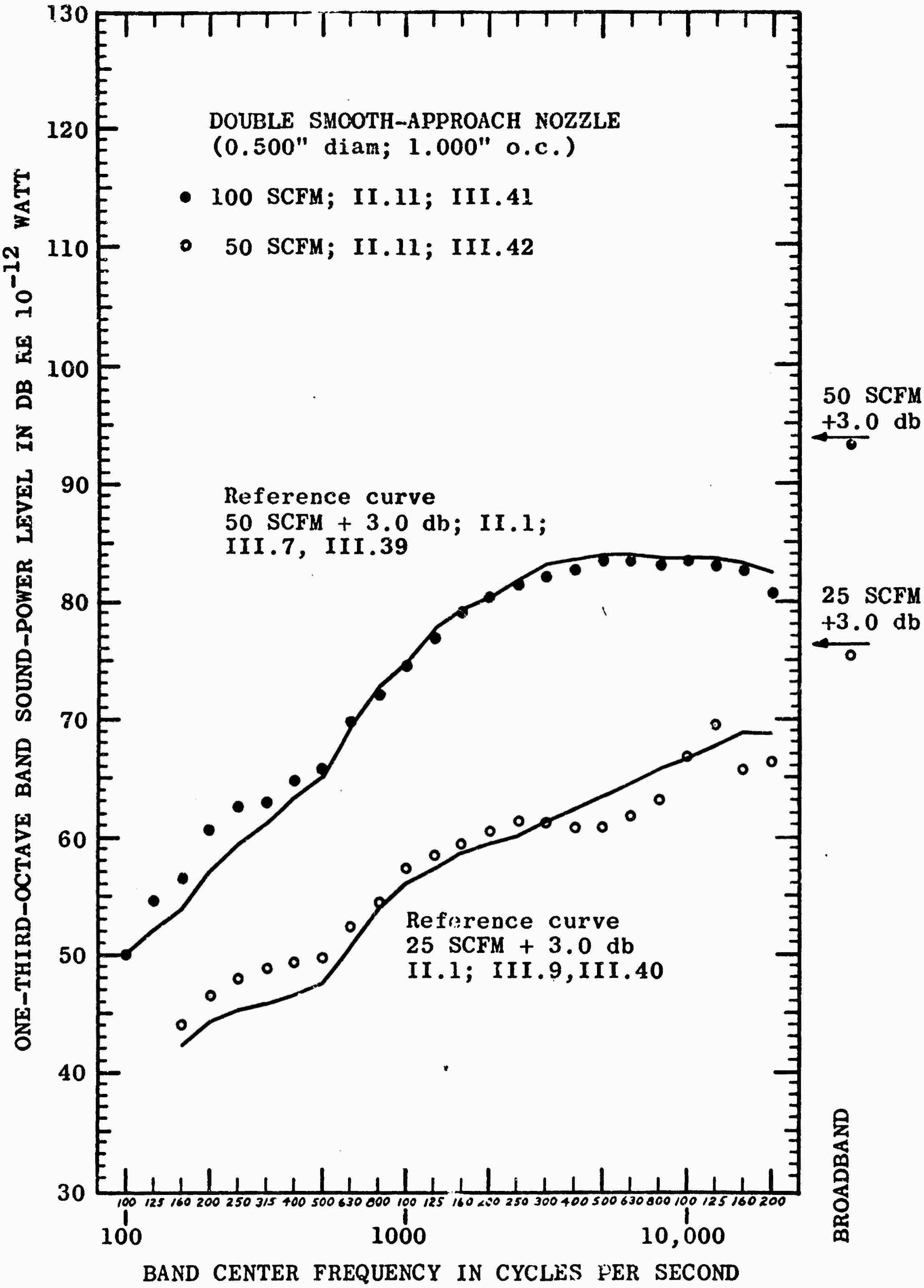

FIGURE 21. TWO 0.500 INCH DIAM NOZZLES 1.000 INCH ON CENTERS.

Figure 20. A number of small changes seem to have occurred but correct interpretation of them is doubtful.

Generally, these limited data for double nozzles are consistent with interpreting the jets as substantially independent noise sources. Small inconsistencies are evident but they are too small and too uncertain to provide a basis for silencer development. Probably some additional studies along these lines ought to be conducted with more favorable flow conditions, say 200 SCFM, and a greater multiplicity of nozzles in order to eliminate some residual uncertainty.

4.4. EXTENSIONS TO NOZZLES

Some tubular extensions were included in these investigations because similar configurations might be needed to connect a jet engine to a muffler and consequently, the acoustical ramifications need to be known. Figure 22 displays the acoustical results of extending a smooth-approach nozzle by more than 95 diameters; see Figure II.13A. Brass tubing with a 0.750-inch internal diameter was selected because it was readily available. Since its diameter was somewhat different from that of any of the smooth-approach nozzles tested previously, a new nozzle, nozzle II.12, was machined to serve as an appropriate reference. One would expect the flow from the reference nozzle to be initially free of turbulence, except of course, for a very small boundary layer. In contrast, the flow from the long extension, considering the applicable magnitude of Reynolds number, should be fully turbulent. At 100 SCFM, the spectrum for the long extension is almost identical to that for the reference nozzle. At 50 SCFM, the long extension produces a spectrum having more nearly the theoretical shape than does the reference nozzle. The observed difference in broadband level is 22.5 db between 100 SCFM and 50 SCFM compared to an expected difference of 24 db if the long extension behaved like a simple jet. Apparently the radius of approach of the reference smooth-approach nozzle is slightly too small for the 50 SCFM flow condition as evidenced by the rising trend in the high-frequency portion of the spectrum.

On the basis of spectrum shape and difference in level with flow condition, this nozzle II.13A with the long extension tube produces results most nearly like those expected for simple jets. This is especially true for the lower velocity flows. Apparently also, the extension tube does not radiate much sound from its outer surface because its spectrum shows no significant increases above the reference spectrum.

The original plan was to successively halve the length of the extension tube and to follow the changes in radiated sound power. However, since at 100 SCFM, the long tube and the reference nozzle gave almost identical results, these investigations were discontinued after testing one other length of extension, namely 36 inches long as given in II.13B. The corre-

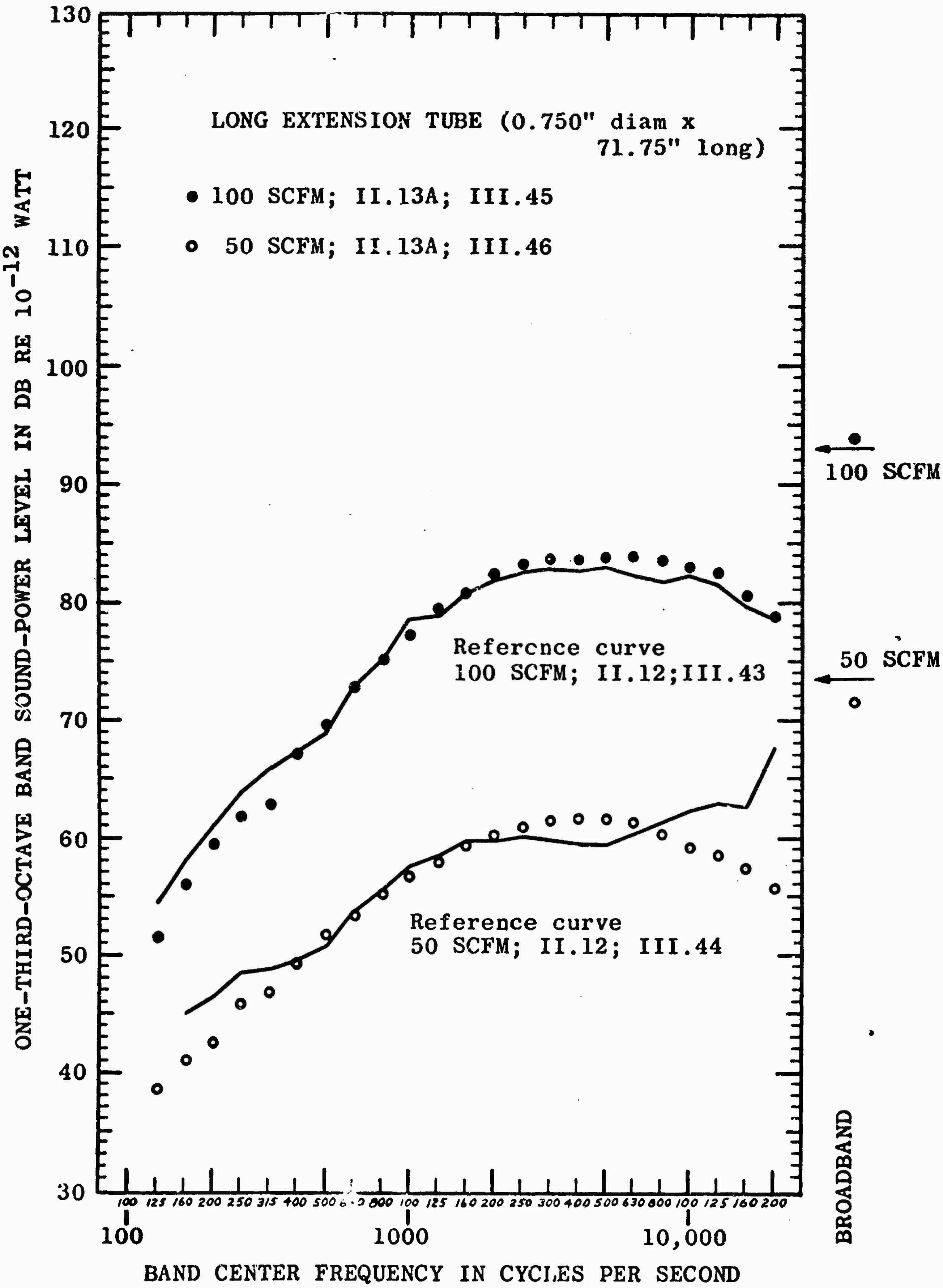

FIGURE 22. EFFECT OF LONG EXTENSION TUBE ON POWER SPECTRA.

sponding spectral data are given in Tables III.47 and III.48 but have not been plotted since they are not sensibly different from the data for the longer extension.

Configuration II.14 is a flattened nozzle which was included in this investigation as a spot check to determine if a larger cross-sectional periphery-to-area ratio had significant acoustical consequences. The area of this nozzle was only about 0.8 that of a 0.500-inch diameter smooth-approach nozzle. Consequently the largest flow rate of 100 SCFM produced a choked flow condition.

Nothing very significant could be deduced from the data (III.51 and III.52) about the acoustical effects of changing periphery-to-area ratio. Spectral shape and level were similar to those expected for simple jets. However, an unexpected phenomenon was observed during the testing of configuration II.14 at large flow rates. The flattened part of the nozzle was forced to vibrate violently by the air flow. The vibration was visible and obvious to the touch. The acoustical result was a large increase in radiated power in the 2500 cps band, as illustrated in Figure 23.

It was not clearly established whether the increased power was radiated directly from the vibrating brass surfaces or indirectly as a consequence of flow modulation. Because of the small area of brass tubing vibrating, the author speculates that the latter explanation predominates. A small machinist's clamp was placed as indicated in II.14 to stop the vibration and the open circles in Figure 23 show the acoustical result. This was the only case detected in which the vibration of a surface associated with the test configuration produced obvious acoustical consequences.

A short extension tube, Figure II.15, was designed to assist the testing of certain muffler configurations; to displace the nozzle exit away from the settling chamber to a position where it would be more accessible. Figure 24 displays the acoustical results in comparison with the 0.500-inch diameter smooth-approach nozzle alone. At sonic flow velocity, the short extension has negligible acoustical consequences. At lower velocities, the extension generates more noise but does not change the spectral distribution very much. The effects are largest at 50 SCFM. In any event, these data for the short extension tube will constitute the reference spectra for the test configurations utilizing the short extension.

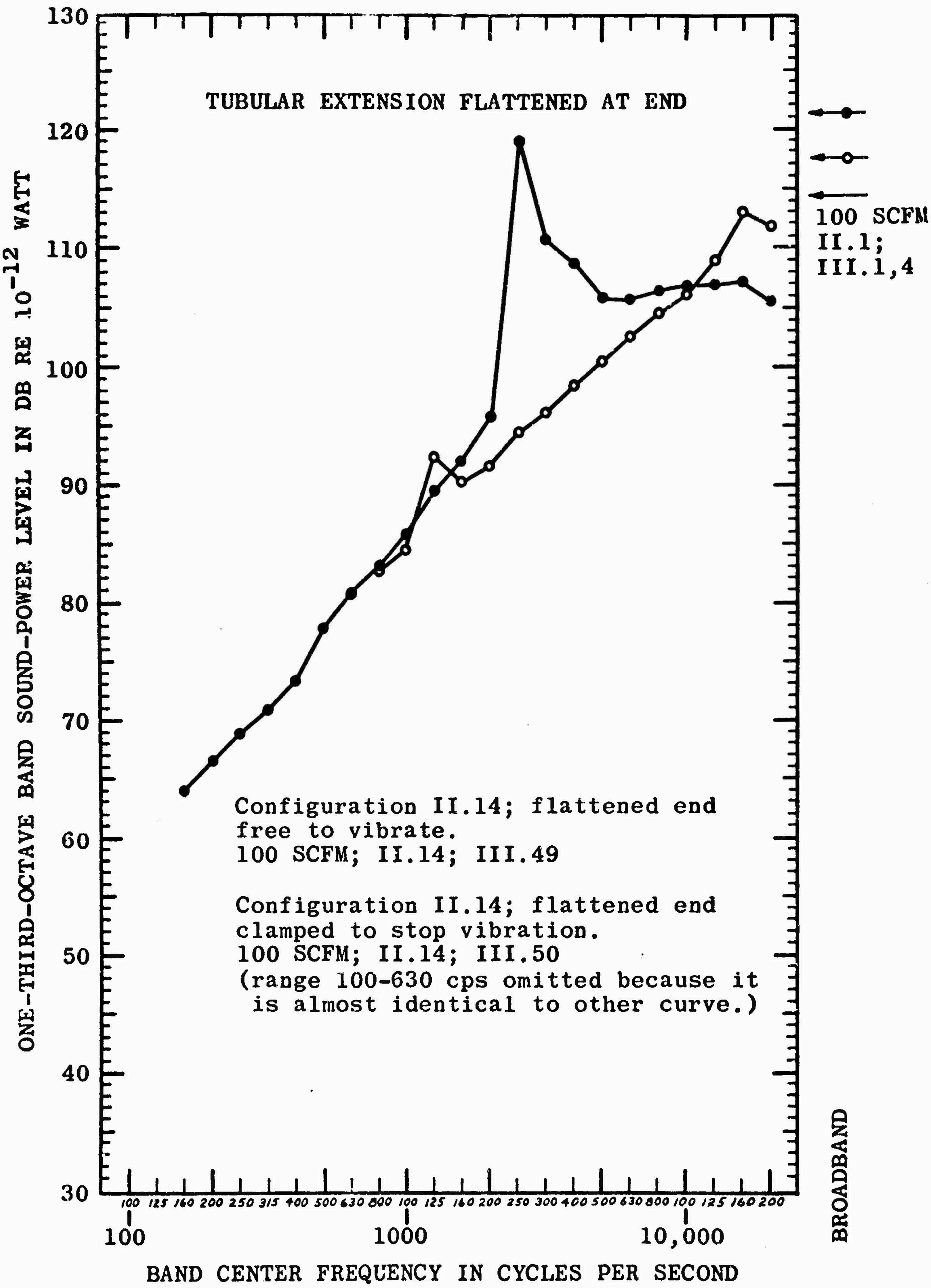

FIGURE 23. ACOUSTICAL EFFECTS DUE TO NOZZLE VIBRATION.

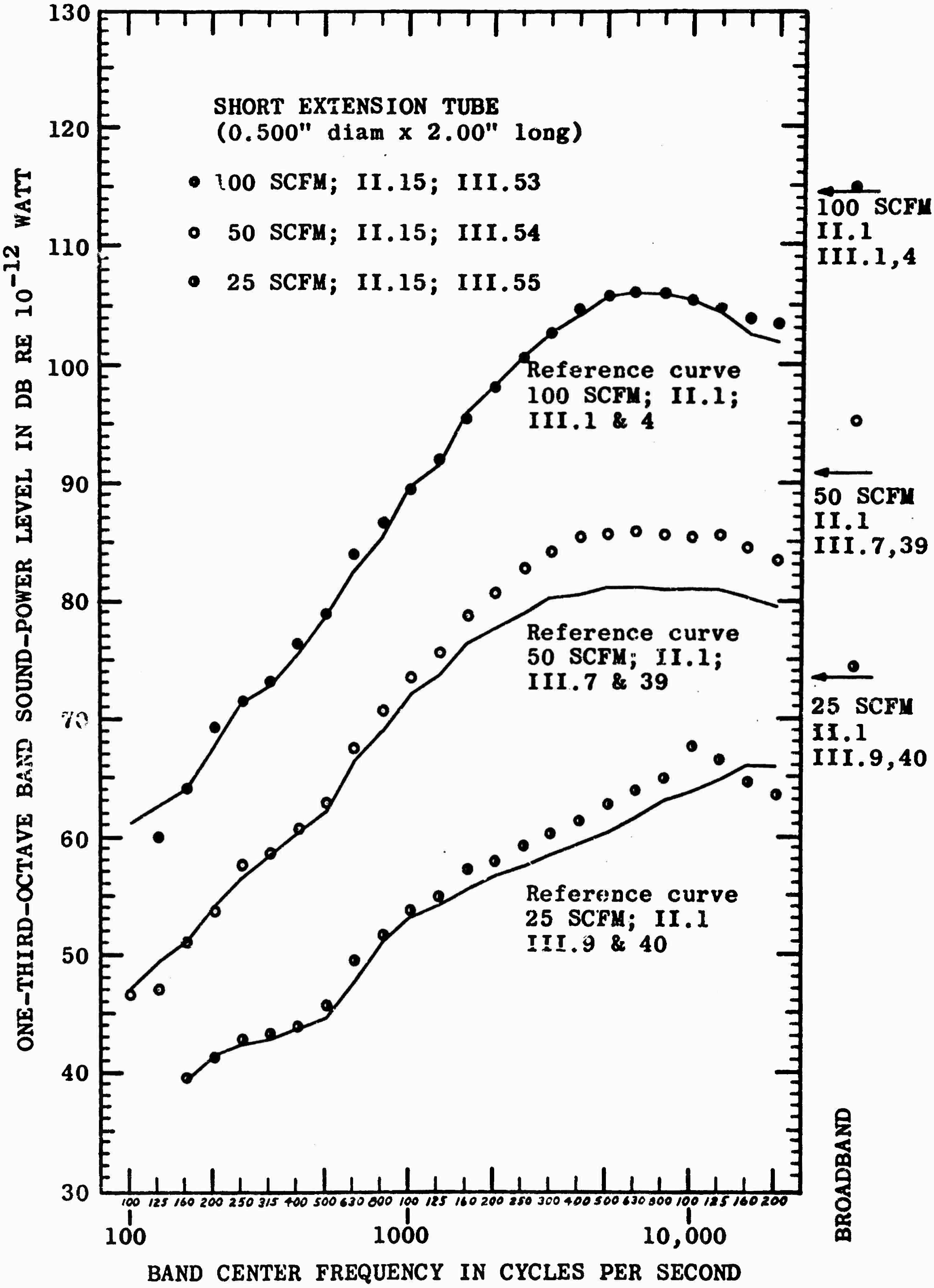

FIGURE 24. EFFECT OF SHORT EXTENSION TUBE ON POWER SPECTRA.

SECTION 5

EXPLORATIONS OF EFFECTS FROM ADDITIONS TO JETS

The experiments reported in this section are concerned mainly with the acoustical results of placing objects in or around the jet exhaust from one or another of the nozzles discussed previously. Thus the experiments are addressed to the question of how to influence the acoustical radiation from a jet once the flow has left the physical constraint of the nozzle. The experiments deal with objects one at a time; composite arrangements of such objects for greater effect are relegated to Section 6.

This section reflects the exploratory nature of the experiments which are not even remotely exhaustive with respect to either the types of objects tested or the ranges of parameters encompassed. Published acoustical literature offers very little concrete guidance except for some mention of screens (see References 4, 16, 17, 18, and 19). One can not even predict with certainty in most cases if a particular object placed a particular way with respect to the jet flow will increase or decrease the radiated power, let alone the effect on spectral distribution. It has been necessary to proceed with little more than intuition for initial guidance.

5.1 TUBES

Several experiments were conducted with sections of tubing surrounding the jet exhaust in an ejector configurate as shown in Figure II.16. A theoretical prediction of the silencing to be expected as a consequence of induced flow is to be found in the literature. (See Reference 5, 662-664, and Reference 20.) Presumably, an exhaust ejector configuration might be encompassed by this theory.

The acoustical data for our experiments are given in Tables III.58 through III.85. Two different lengths of tube were used and these were positioned at various distances from nozzle II.1. The principal results are displayed in Figures 25 through 28. The most obvious results are: (1) more noise is generated with the tubes present than by the simple nozzle alone, and (2) tonal generation occurs.

A more detailed analysis of these graphs and data reveals additional interesting results. In Figures 25 through 28, the solid dots refer to a test configuration in which the tubes were positioned tightly against the nozzle face. This configuration resembles the closed-pipe discussed in all elementary physics texts and, of course, ejector action is impossible since no passage exists for admitting secondary air flow. For such a pipe, closed

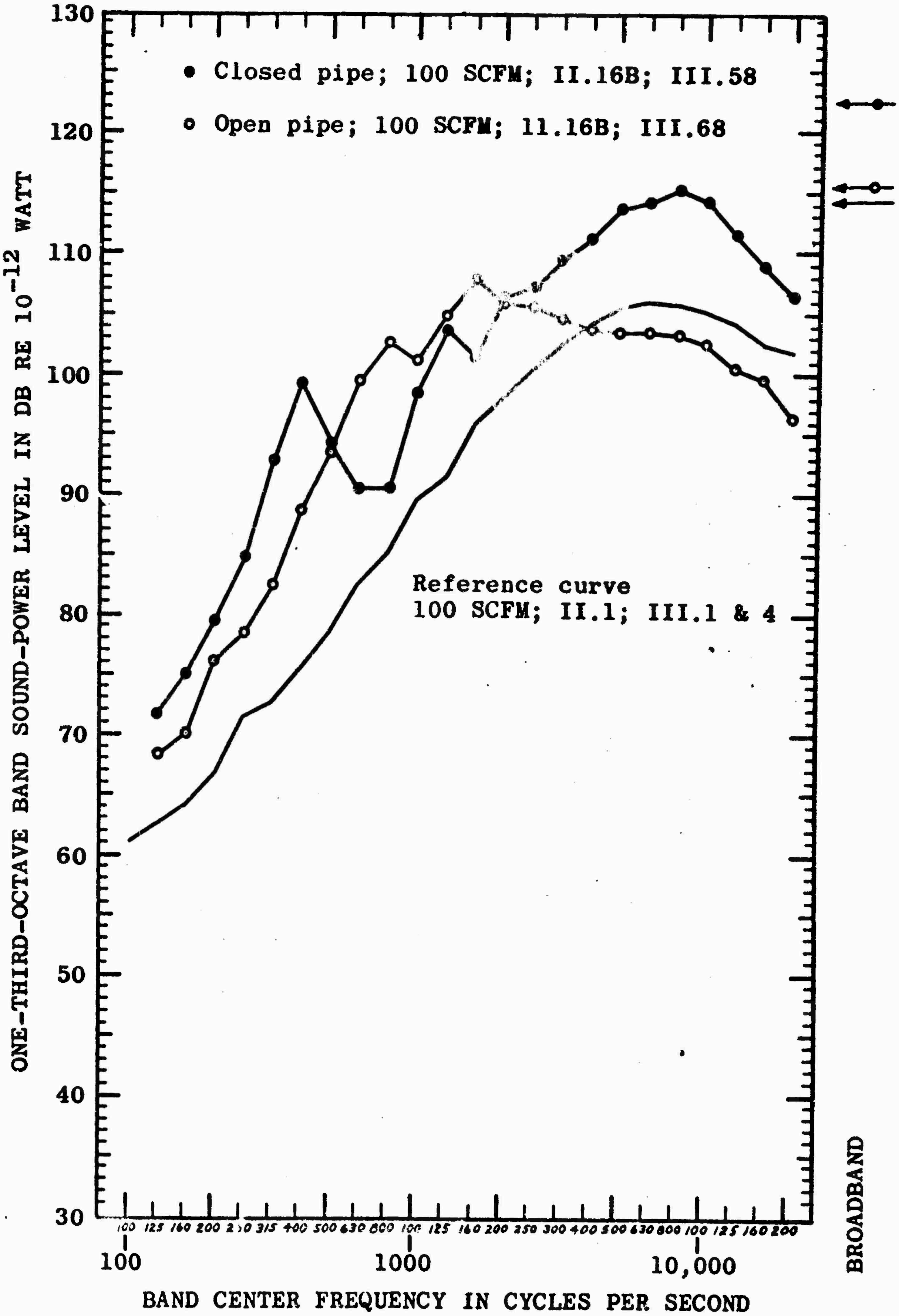

FIGURE 25. TUBE 8 INCHES LONG OPERATED AT 100 SCFM FLOW RATE.

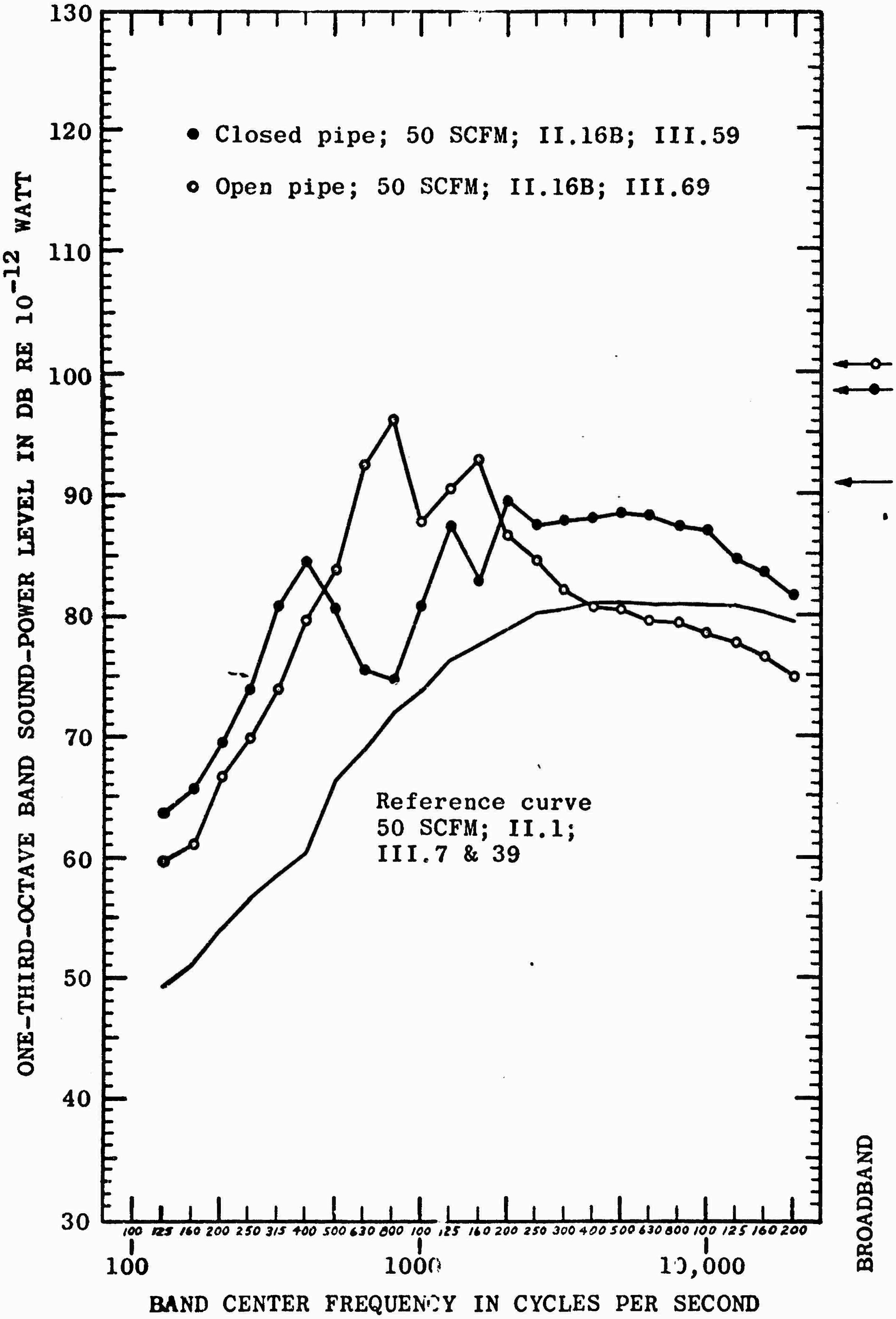

FIGURE 26. TUBE 8 INCHES LONG OPERATED AT 50 SCFM FLOW RATE.

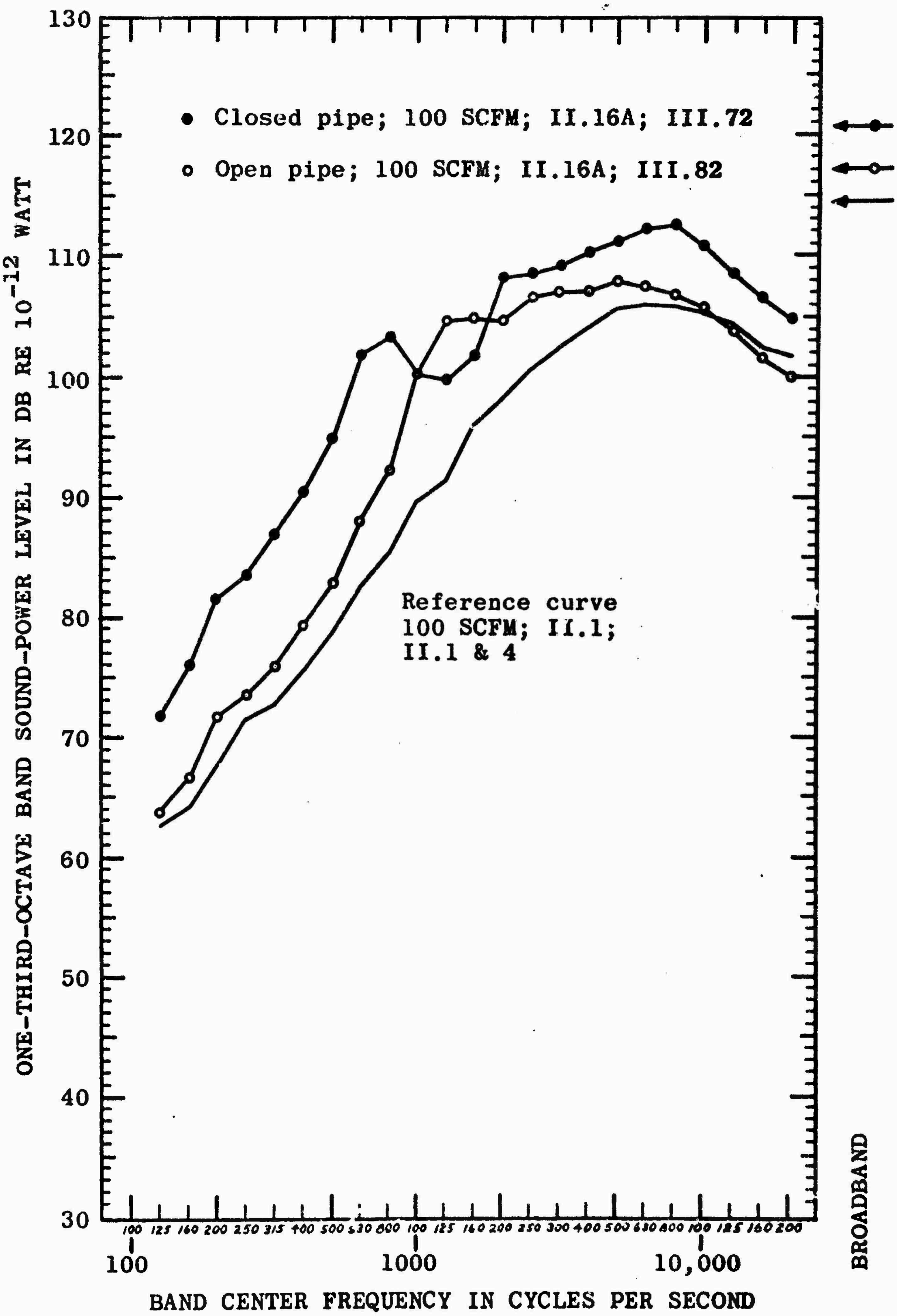

FIGURE 27. TUBE 4 INCHES LONG OPERATED AT 100 SCFM FLOW RATE.

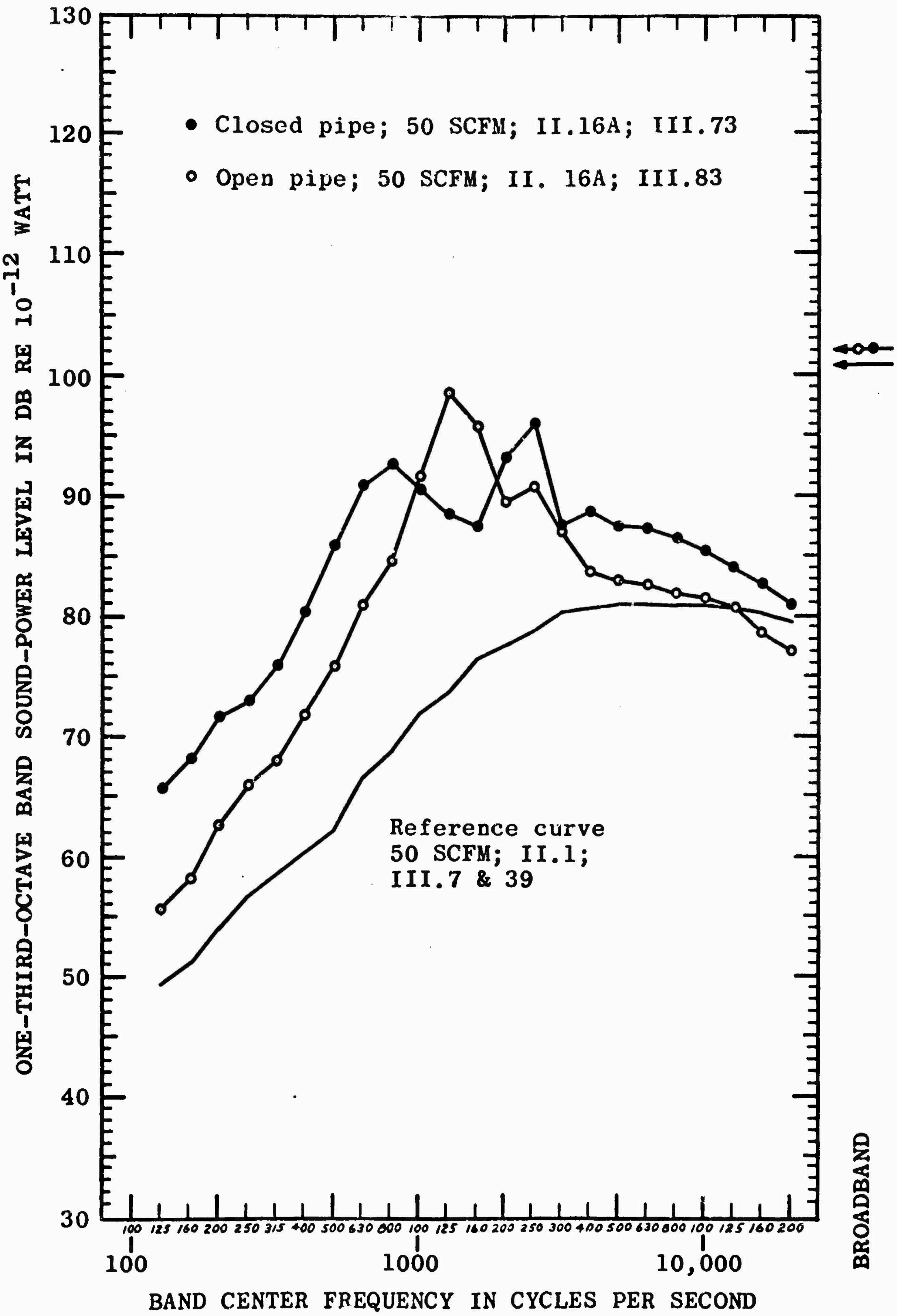

FIGURE 28. TUBE 4 INCHES LONG OPERATED AT 50 SCFM FLOW RATE.

at one end and open at the other, the fundamental resonant frequency (see, for example, Reference 21) will be:

$$f_1 = \frac{c}{4l}$$

where f_1 = fundamental frequency

c = velocity of sound

l = length of the pipe

The effective acoustical length of a pipe is slightly longer than its physical length. The correction to be made is of the order of 0.8 times the radius of the pipe. Assuming the velocity of sound to be 1130 feet per second, the above formula predicts

$f_1 \cong 400$ cps for the 8-inch long closed tube

$f_1 \cong 725$ cps for the 4-inch long closed tube

Thus for the 8-inch long closed tube, one would expect to find evidence of the fundamental frequency in the 400 cps band and Figures 25 and 26 verify this prediction. Likewise, for the 4-inch long tube, one would expect to find a maximum in the 800 cps band or perhaps shared by the 630 and 800 cps band; a prediction which Figures 27 and 28 confirm.

For a closed pipe, the overtones occur at odd harmonics of the fundamental; that is, $f_2 = 3f_1$, $f_3 = 5f_1$, etc. In a one-third octave band spectrum, f_2 would appear in the fifth band above the fundamental, f_3 in the seventh band, and higher overtones cannot be resolved. This predicted behavior likewise is confirmed in Figures 25-28. Theory has not succeeded in predicting the intensity of the tonal generation and so the experimental data alone provide evidence of magnitudes.

The open circle dots displayed in Figures 25-28 all represent an x/d 2 test configuration. Secondary air flow is permitted and ejector action occurs. This configuration resembles an acoustical pipe open at both ends. Theory predicts a fundamental frequency of (see Reference 21):

$$f_1 = \frac{c}{2l}$$

Proceeding as before and making some adjustment for end corrections, we predict:

$$f_1 \cong 725 \text{ cps for the 8-inch long open tube}$$

$$f_1 \cong 1250 \text{ cps for the 4-inch long open tube}$$

These values are consistent with the local maxima evident in Figures 25-28. Furthermore, the overtones of an open pipe include all harmonics of the fundamental. Therefore, the second harmonic should appear three bands above the fundamental, the third harmonic five bands above the fundamental, and higher overtones will remain unresolved. In Figures 25-28, evidence of the second harmonic appears where predicted. The third harmonic is too weak to be observed; perhaps with sufficient imagination it can be located in Figures 25 and 26.

Although Figures 25 through 28 display only two locations of the tubes, namely, $x/d = 0$ and $x/d = 2$, five other locations, $x/d = 1/8, 1/4, 1/2, 1,$ and 4, were tested and all of the acoustic data are in Appendix III. The $x/d = 2$ condition was selected as representative of the open pipe configuration. All of the other spacings yield comparable results except, perhaps $x/d = 1/8$; in this case, the inlet aperture for secondary air is so narrow that it is questionable whether the configuration is more like a closed pipe than an open pipe. The local maxima are poorly defined but the magnitude of the sound power is similar band for band with the other conditions tested. Actually, the 50 SCFM test condition displays the tonal characteristics more prominently than the 100 SCFM condition for almost all spacings tested.

When the acoustical source conditions are as complex as those applying to these tests with tubes, it is not very practicable to consider total or broadband power as the sole measure of acoustical performance. The difficulty is illustrated in Figure 29 where broadband sound power level is plotted against x/d with flow rate as parameter. These plots trend in different directions which effectively frustrates attempts at generalization.

If the concept of silencing jet noise by induced flow (see References 5 and 20) can be applied to ejectors, and Reference 5 states that it may apply, a sound power reduction of 37 db would be predicted for our test configuration. This prediction is based on an area ratio of 16 and cold jet behavior. To the contrary, all of the present experiments have demonstrated an increase of radiated sound power. The increases have been due principally to enhanced low-frequency radiation and to tonal generation; the open-pipe data have shown some moderate decreases at high frequencies. It is evident that tonal generation, which is the normal acoustical characteristic of pipes, has not been taken into account in the theory pertaining to induced flow. Thus, when considered in conjunction with ejector configurations, the theory of induced flow would seem relegated to the role of a

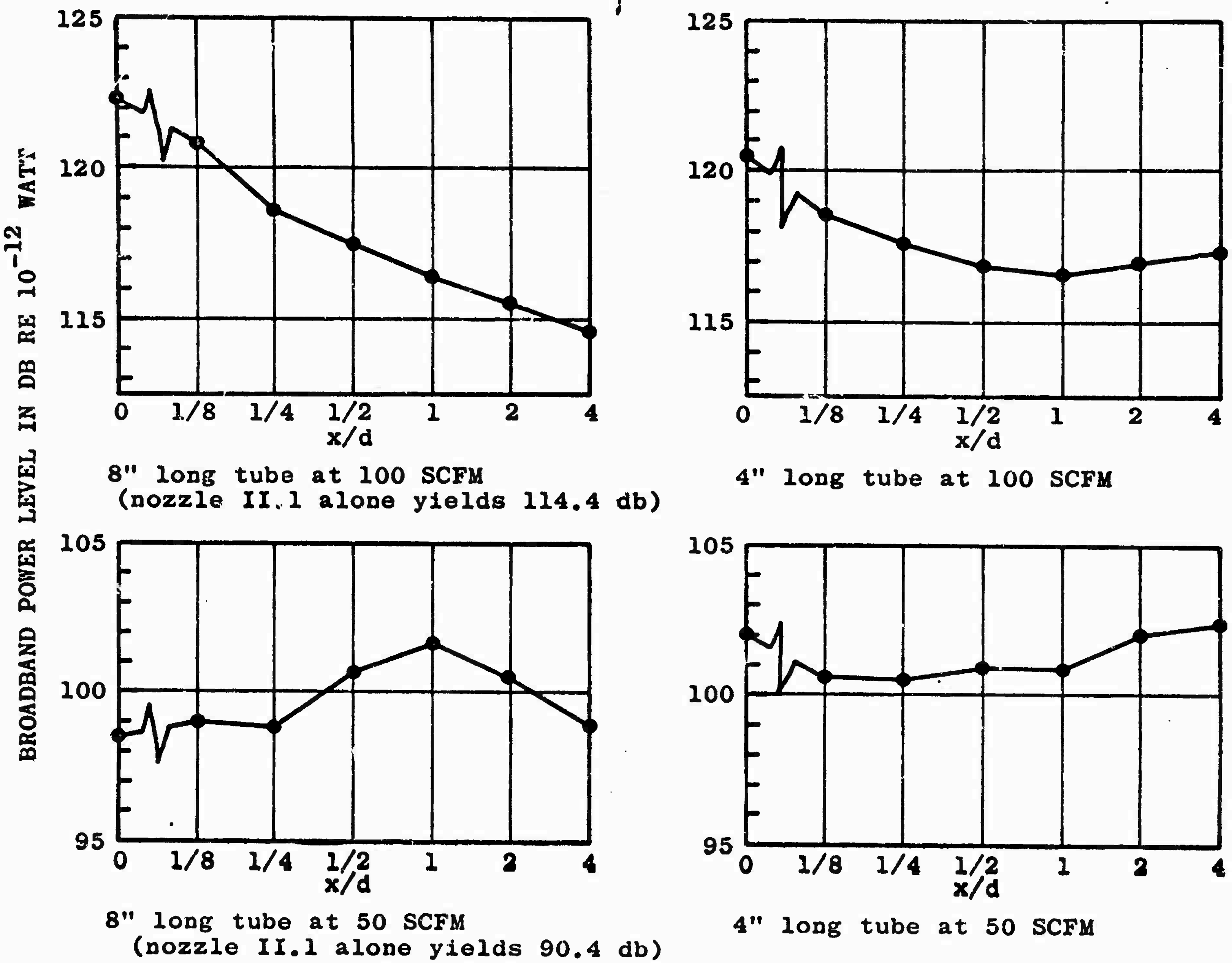

FIGURE 29. BROADBAND POWER ASSOCIATED WITH TESTS ON TUBES.

lower limiting case. Perhaps the silencing predicted from the temperature change accompanying hot jet operation would be easier to realize than that due to area in the case of an ejector configuration.

The present experiments indicate a somewhat more involved description of the effects of tubes on jet noise. First, the radiated sound power is increased by the presence of a tube surrounding the jet exhaust. The effect is most pronounced at low and medium frequencies; at high frequencies the behavior is not so clearly delineated The closed pipe configuration is still noisier than the bare nozzle through 20 kc, but the open pipe configuration hints at reduction; perhaps due to shielding, perhaps due to induced flow lowering the power and shifting the peak to lower frequencies for the simple jet issuing from the ejector tube. Secondly, there is a tonal characteristic to the radiated power which correlates with the normal mode structure appropriate to the boundary condition imposed by the tube configuration. The antiresonances or valleys between the resonant peaks do not dip as far as the level of the nozzle alone.

It appears likely that the ejector configuration yields:

(1) that portion of the bare nozzle's spectrum which is radiated from the jet before it enters the ejector tube,

(2) probably a simple jet spectrum characteristic of the velocity and area at the exit from the ejector,

(3) increased radiation due to the presence of solid boundaries in or near the jet flow,

(4) a tonal characteristic superimposed upon the above.

In the present experiments, the tube walls were massive, rigid, and damped by the clamp supporting the tube. Flexural resonances of the tube itself do not seem to have influenced the acoustical result.

By hindsight, it seems as if increased acoustic radiation should have been the anticipated result because:

(1) no mechanism for diminishing the efficiency of noise generation was in evidence; to the contrary, more "radiating" surface was introduced.

(2) no dissipative mechanism was included, such as an absorptive lining, to absorb some of the sound energy once it was generated.

The experiments reported in detail here have utilized only one diameter of tube. Other less detailed investigations were conducted with tubes ranging from about three-quarters inch to four inches in diameter. These ranged

from only an inch or two long to several feet long Some of the short larger-diameter tubes had a geometry such that the expanding jet exhaust from the nozzle did not impinge the tube walls. All of these configurations tended toward increased noise similar to that already detailed. Several experiments were tried using nozzle II.13 so that the surrounding tube could be moved completely upstream of the nozzle exit. As the surrounding tube was shifted downstream, increased noise became evident when the surrounding tube began to enclose the jet exhaust.

One additional observation on the subject of tubes is worth mentioning perhaps. A section of tubing about 8 inches inside diameter, 8 inches long, 0.8 inch thick made of compressed glass fibers was used for an experiment.[11] It was hand held around the jet from nozzle II.1 and its dimensions were so large that the expanding jet did not impinge on it. The occurrence which surprised the author was the pronounced buffeting caused by the induced flow. It was definitely difficult to hold this fiberglass tube centered around the jet flow. The buffeting forces determined by tactual sense were of very low frequency but lent credulity to the increased sound power at audible frequencies. The original intent was to investigate the ability of the somewhat permeable fiberglass tube to contain and attenuate radiated noise. A thick permeable partition is known to provide more than mass-law attenuation at high frequencies where the wavelength of sound is comparable to partition thickness. (See Reference 22) This particular experiment was one of those never accomplished for lack of time.

The investigation of tubes reported here has really just introduced the subject. Much more research is needed to explore fully the acoustical ramifications in a quantitative manner. Future experiments should probably be planned so that the amount and form of the induced flows can be entered as controlled parameters.

5.2. PLATES

Several experiments were conducted to investigate the acoustical consequences of deflecting the jet flow with a solid plate placed either normal to the flow as in Figure II.17 or at an angle to the flow as in Figure II.18. Geometrically, these configurations contrast with those of the previous section. In this instance, an extended surface is merely exposed in the flow whereas previously, it surrounded the flow. The experimental data are contained in Tables III.86 through III.99. Figure 30 displays the acoustical results at 100 SCFM when the plate is normal to the jet with x/d as parameter. A series of weak broad maxima and minima appear to be superimposed onto the curve for

[11] Actually 8 inch G-B duct with the outer vinyl covering removed. Manufactured by the Gustin-Bacon Manufacturing Company, Kansas City, Missouri.

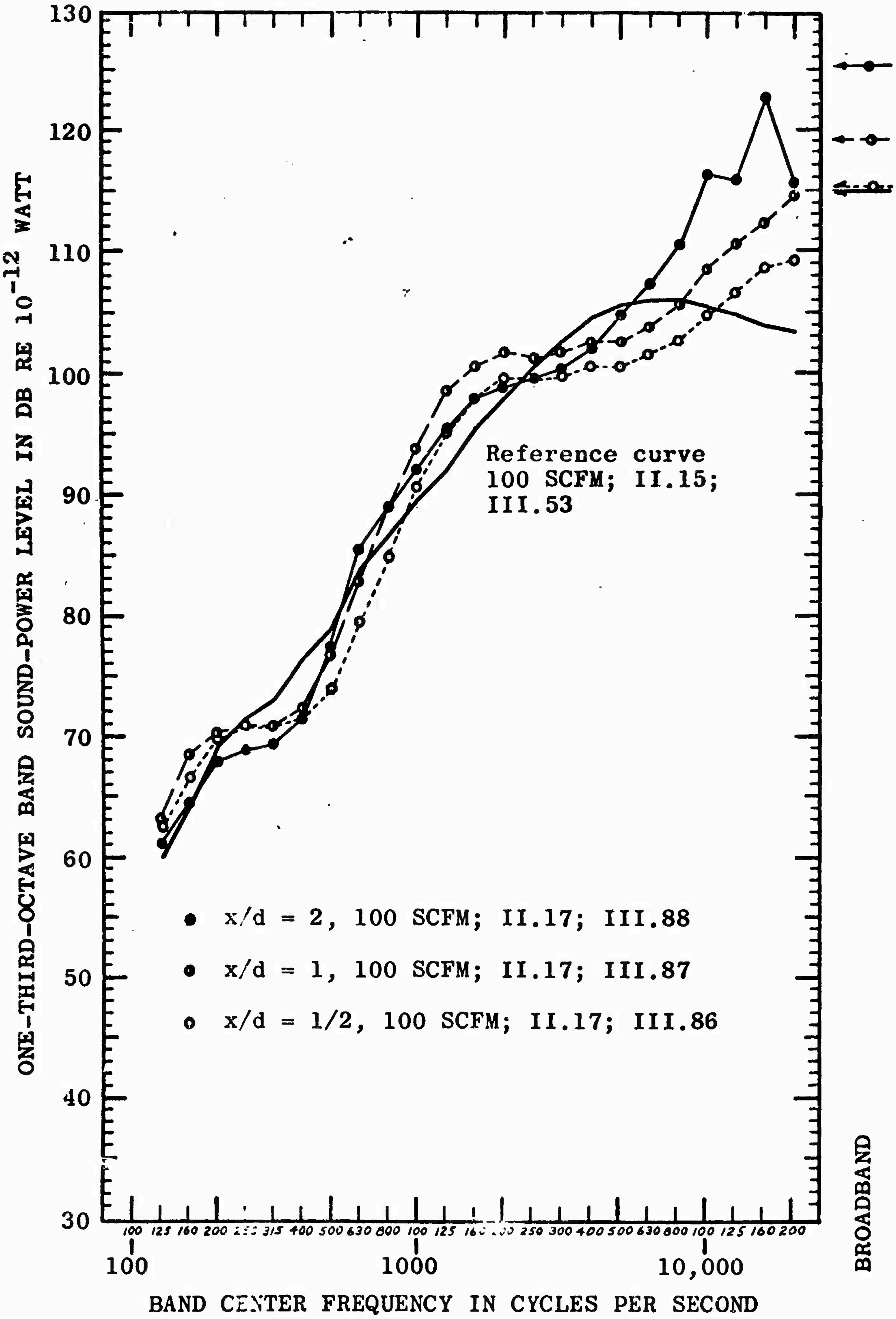

FIGURE 30. PLATE NORMAL TO JET AXIS AT 100 SCFM.

the nozzle alone. The first maximum appears at about 200 cps and the next around 1600 to 2000 cps. These two maxima seem to be essentially independent of the distance of the plate from the nozzle. At the upper frequency limit of 20kc, an increasing trend again becomes evident and here the results are ordered with respect to the distance parameter x/d, larger spacing causing a shift toward lower frequencies. The broadband sound power is increased by the presence of the plate, chiefly due to the high-frequency contributions. The high-frequency phenomenon does not appear to have been completely displayed in this experiment.

Figure 31 displays the acoustical results for a plate normal to the jet axis and a flow rate of 50 SCFM. Now two maxima are evident and they appear as additions to the bare nozzle reference curve. The lower frequency maxima near 1250 cps appear almost independent of x/d while the high-frequency maxima are ordered inversely with respect to x/d.

It was difficult to obtain a rigid vibrationless support of the plate but vibration of the plate-support arrangement is not thought to be responsible for the acoustic effects noted. The maxima in the vicinity of 1250 cps occur in a range where the lateral extension of the plate, 4 inches, is of the order of 1/4 to 1/2 wavelength. Further experiments will be needed to determine if plate dimension is a significant parameter.

When the plate is tilted at 45° with respect to the jet's axis, the acoustical results change in appearance as Figures 32 and 33 demonstrate. Data were collected for four values of the parameter x/d but only that for x/d = 0 and x/d = 2 have been plotted because the results are so nearly alike. There appears to be a small but real reduction in the low-frequency noise and a moderate increase in the high-frequency noise. As usual, the more pronounced effects are observed at the lower flow rate of 50 SCFM.

The results for flat plates have not been as dramatic as those obtained with tubes. Neverthless, in all cases tested, the broadband power had been increased above that for the bare nozzle. In some instances, the increases were moderate so that conceivably, when used in conjunction with highly effective silencing configurations, these increases might be tolerable if deflection of the jet flow were desirable.

5.3. SCREENS

Screens placed transversely a short distance downstream from a nozzle have been tried as noise reduction devices. Early experiments (see References 4, 16, 18) clearly demonstrated that, although the loss of thrust prohibited the use of screens during flight, screens did produce enough acoustical effect to suggest their potential application in ground silencer designs. Those experiments further demonstrated that the acoustical effects varied with distance from the nozzle, that the directionality pattern was altered by the

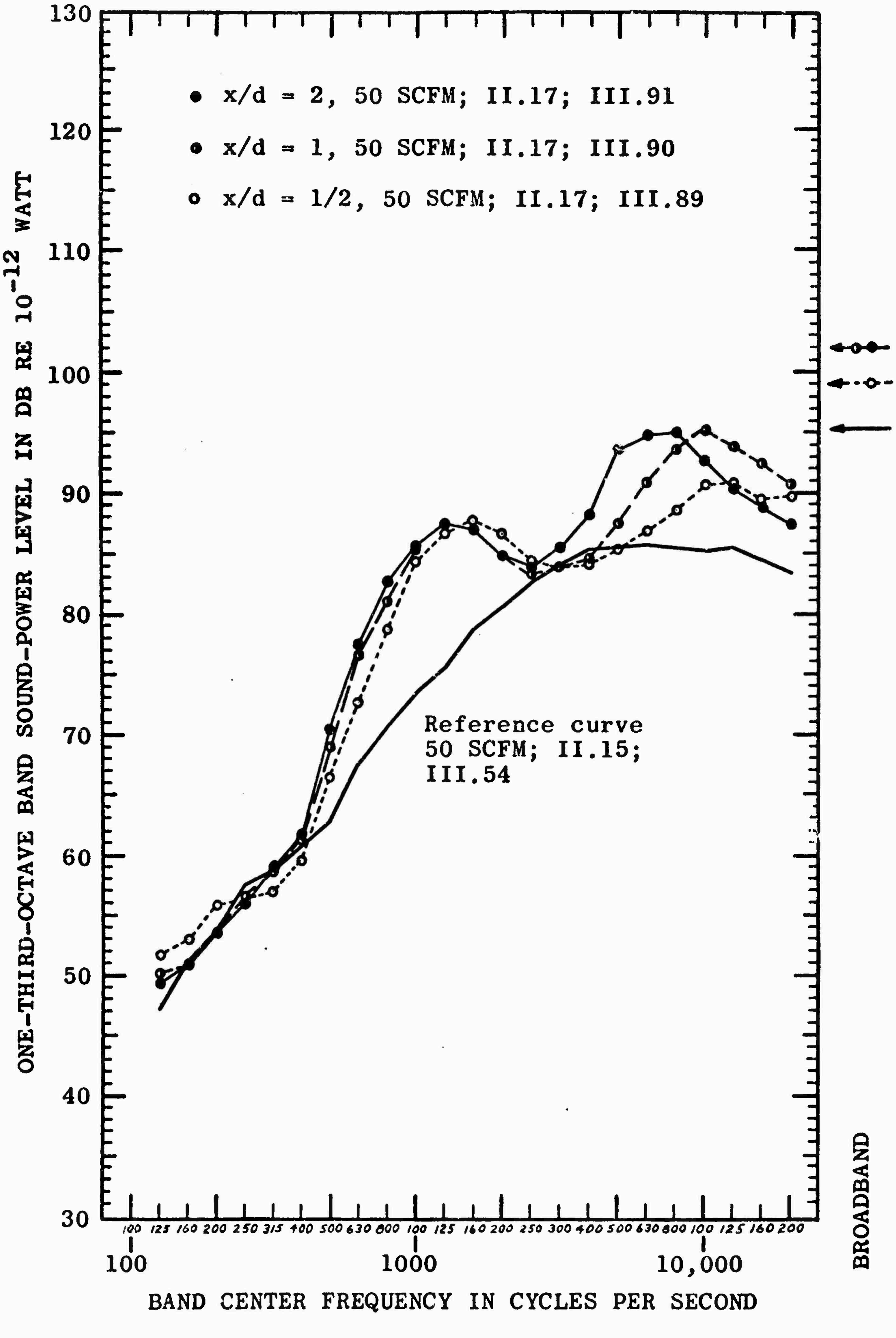

FIGURE 31. PLATE NORMAL TO JET AXIS AT 50 SCFM.

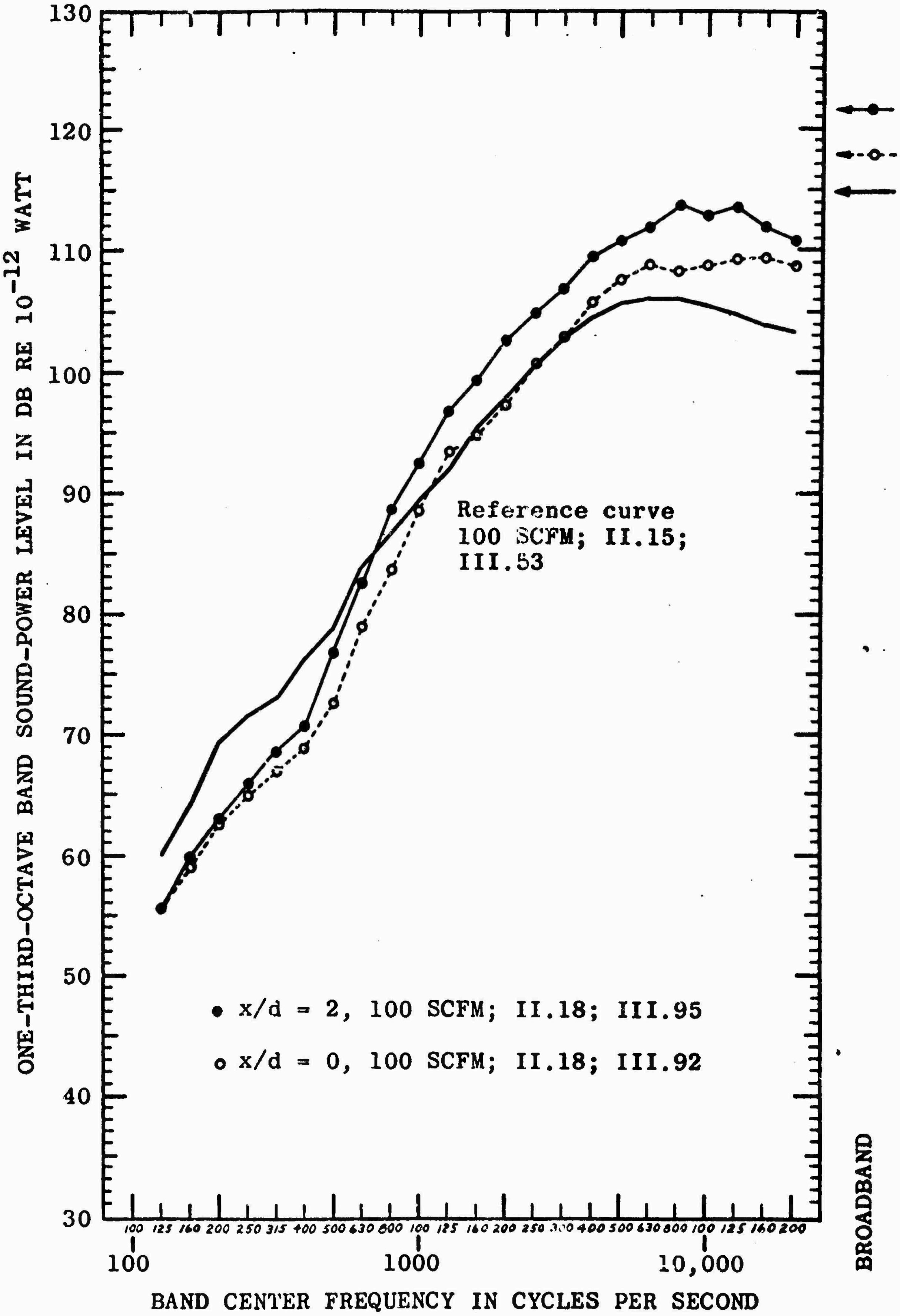

FIGURE 32. PLATE AT ANGLE TO JET AXIS AT 100 SCFM.

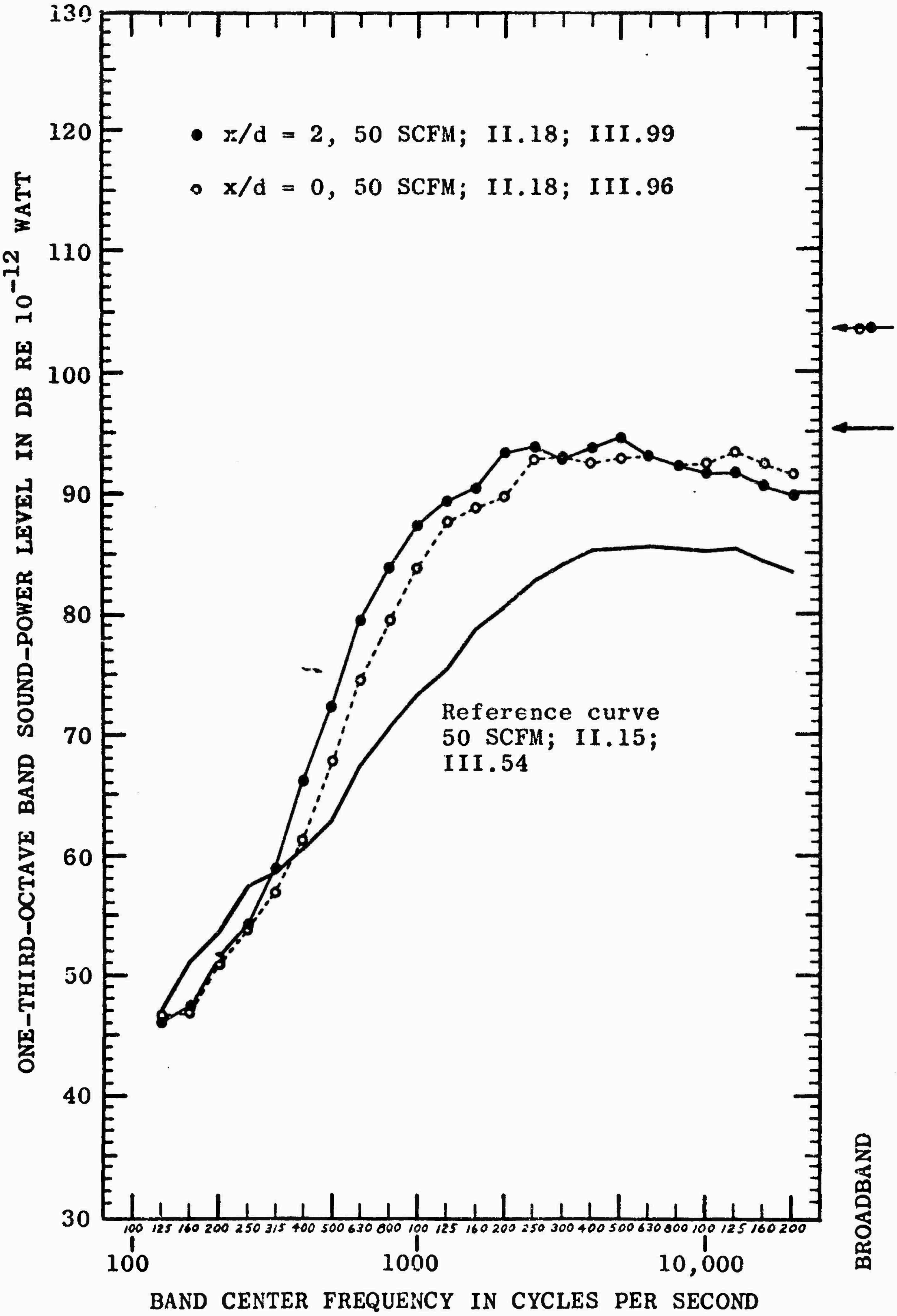

FIGURE 33. PLATE AT ANGLE TO JET AXIS AT 50 SCFM.

screens and indicated that finer screens contributed more silencing.

Only part of the literature referring to screens as silencing devices was immediately available when planning the experiments reported here. It was apparent, nevertheless, that rather limited ranges of the various possible parameters concerning screens had been encompassed and that the changes in directionality and spectrum occurring simultaneously were tending to complicate interpretation of the reported free-field measurements. The use of uniform, square-mesh screens woven from round rods and placed normal to the jet axis drastically limits the scope of possible parameter variation but in a manner which does not seem unreasonable. Considered most generally, a screen-like object placed in a jet to influence the acoustical result introduces a fantastic array of parameters to be investigated. Among these parameters are:

1. The location of the screen downstream from the nozzle exit,
2. the mesh of the screen relative to a given nozzle diameter,
3. the percentage of open area in the screen, i.e., the wire size relative to the mesh,
4. the flow velocity for a given nozzle diameter,
5. the cross-sectional shape of the wire,
6. the type of weave used to fabricate the screen,
7. uniform vs. non-uniform distribution of mesh and/or wire size,
8. scaling as evidenced by changing nozzle diameter,
9. screen placement other than normal to the jet axis,
10. multiple (series) arrangements of screens,
11. use of screens in combination with tubes, plates, etc.,
12. detailed redistribution of the acoustic spectrum rather than just total power considerations.

There remains, of course, the matter of directionality for which the reverberation room is not appropriate and the effects of jet temperature which were intentionally omitted from this research program.

Considering some of the possible parameters mentioned above, if we were to investigate completely three flow velocities for three nozzle diameters at eleven distances downstream for twelve mesh sizes with four different percentages of open area in the 24 one-third-octave bands between 100 and 20,000

cps, more than 100,000 comparisons would be involved. Naturally, such an elaborate investigation was impossible in this research program but many spectra have been collected for a selected variety of parameters. The principal data are recorded in Appendix III but only a few of the results can be presented in this discussion.

Several screens, characterized by different meshes and wire sizes, were selected from the current stock of a local wire cloth manufacturer. It was considered inappropriate in this exploratory research to incur the expense of special fabrication unless and until a definite need had been demonstrated for non-stock items. Because existing literature pointed toward improved results with finer meshes, the initial selection of screens emphasized very fine screens. The screens obtained are listed in Table 3 and the most complete tests with them were accomplished with the one-half inch diameter smooth approach nozzle (Figure II.1) which produced a Mach one (approximately) jet at a flow rate of 100 SCFM.

TABLE 3

CHARACTERISTICS OF EXPERIMENTAL SCREENS

Mesh, wires/inch	Wire diameter, inch	Approximate % open area
10	0.025	56
20	0.013	55
30	0.012	42
40	0.008	46
50	0.009	30
60	0.010	16
60	0.0065	37
80	0.0055	31
100	0.0045	30
120	0.0036	32
200	0.0025	25
200	0.0021	33
300	0.0015	29
400	0.0011	31

Since a full-size jet has a tail-pipe which is about 20 inches in diameter, a linear scaling to a one-half inch diameter model nozzle would be 40:1. On this basis, the coarsest screen (10 mesh) listed above corresponds to a full-scale screen having one-inch diameter bars spaced four inches on centers. The finest screen (400 mesh) scales to 10 mesh with 0.044-inch diameter wires. Thus, applying the 40:1 scaling factor, the screens listed in Table III encompass a somewhat greater range of mesh size than reported in Reference 18

although exact scaling for all parameters does not necessarily exist. Reference 18 also indicated that, everything considered, a screen of one-quarter inch diameter wire spaced one-inch on centers seemed best. That screen corresponds most closely to our 40 mesh screen except that our wire diameter is slightly larger; 0.008 inch instead of 0.006 inch required for exact scaling.

There are so many parameters interacting that it is difficult to know where to start making comparisons; small effects found upon changing one selected parameter may merely be the consequence of inadvertently choosing far-from-optimum values among the other relevant parameters. In order to display the major role of the distance of the screen from the nozzle, the results obtained with the 20 mesh screen are illustrated in Figures 34 and 35. When the screen is placed tightly against the nozzle ($x/d = 0$), it naturally raises the pressure needed to maintain the same mass flow rate. However, even under this condition, a significant reduction in sound power was observed at all frequencies within our measurable-frequency range. The silencing was most pronounced in the vicinity of the spectral maximum for the bare nozzle. Moreover, the silencing effect was considerably larger for a jet of high initial velocity (see Figure 34; 100 SCFM corresponds roughly to Mach one for the unobstructed nozzle) than for a jet of moderate initial velocity (see Figure 35; 50 SCFM corresponds roughly to Mach 0.5).

Actually for the conditions prevailing in Figure 35 with the 20 mesh screen placed tightly against the nozzle, the small individual jets from each of the screen openings ought to have mean velocities close to Mach one on the basis of 55% open area. However the measured value of upstream pressure was only 5.2 psig (see Table III.122); a value far below critical pressure. Judging from the observed pressure ratio, this screen acts as if it were about 89% open instead of 55% open. Two factors may account for the apparent discrepancy. First, the screen may have been forced slightly away from the nozzle face which, in consequence, would permit some lateral flow thereby altering the area ratio. Second, the screen has finite thickness due to the weaving of its wires and therefore the flow passages are larger, in microscopic detail than would be predicted on the basis of projected open area. Moreover, the experimental situation is one which may not be adequately represented by the usual elementary isenthropic relationships. Nevertheless, the experimental fact remains that a much larger acoustic change was observed in Figure 34 than in Figure 35. The velocity of the primary flow, or some closely related quantity, therefore appears to be an important parameter.

As the screen is placed farther downstream from the nozzle, its augmentation of the backpressure decreases rapidly. Although our measurements of backpressure were rather insensitive, no augmentation was observed if the screens were placed at $x/d = 1/2$ or greater. A similar result was obtained during experiments with flat solid plates (see Section 5.2) and this result is consistent also with the backpressure situation reported in Reference 18 for full-scale tests with screens.

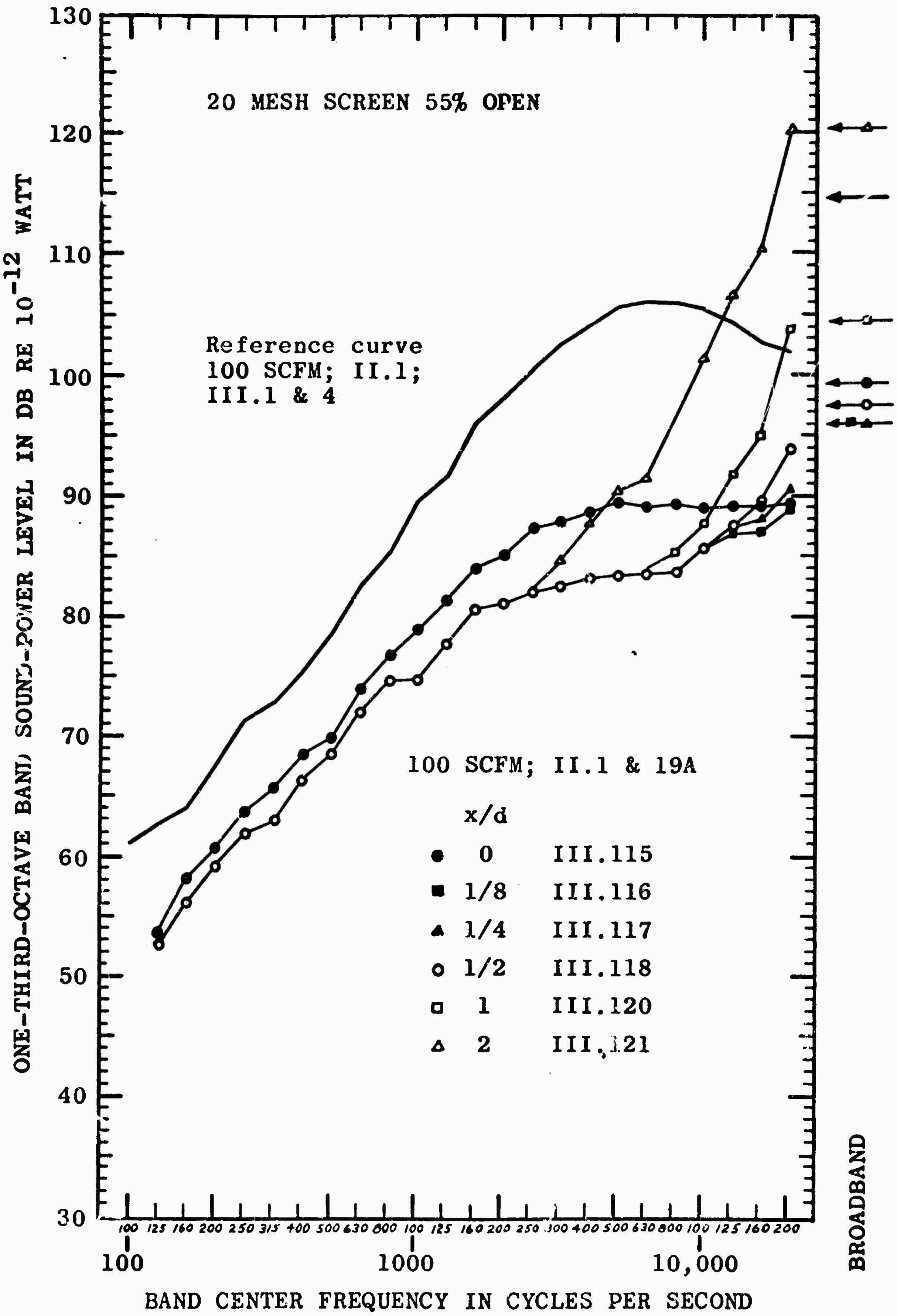

FIGURE 34. EFFECT OF DISTANCE PARAMETER; HIGH VELOCITY.

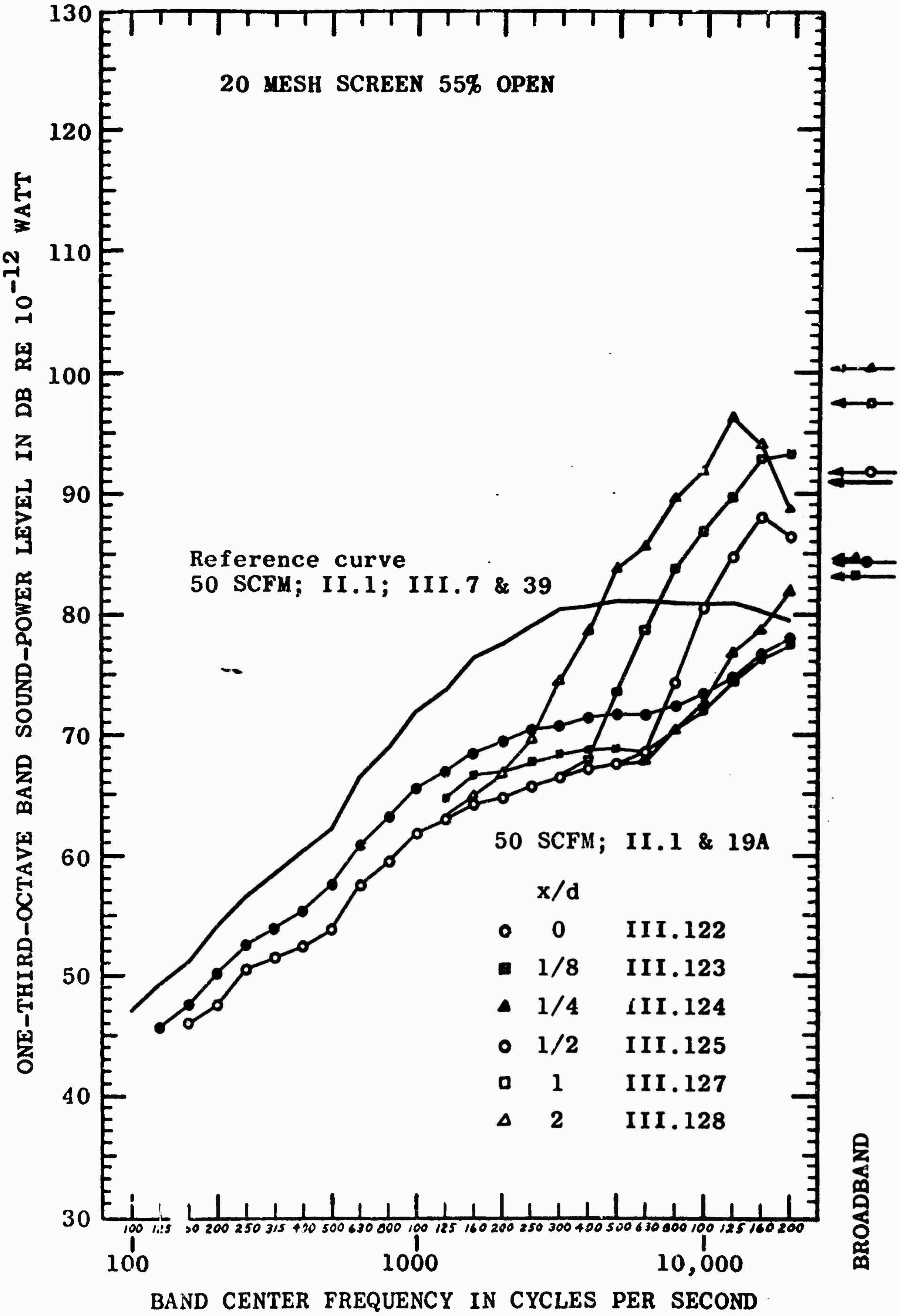

FIGURE 35. EFFECT OF DISTANCE PARAMETER; MEDIUM VELOCITY.

It will probably be noticed that the tables in Appendix III relating to screens include the remarks, "side flow permitted" or "side flow not permitted." "Side flow permitted" refers to a configuration in which the screen was spaced away from the nozzle face by three small cylindrical columns of appropriate length. Lateral air flow, occasioned by a screen, could escape between the nozzle and the screen with this configuration. "Side flow not permitted" refers to a configuration in which the screen was spaced away from the nozzle face by a complete ring of appropriate thickness. In this case, any lateral flow of air ultimately had to change direction again and flow out through the screen. With either configuration, the diameter of the jet efflux could expand as flow conditions through the screen dictated. The existance or prohibition of side flow, in the above sense, had small acoustical consequence but the designation was preserved for explicit description of the experimental configuration. Later on, several experiments are described in which the efflux was limited to a central area on the screens by means of cover plates and those experiments are to be differentiated from the experiments presently discussed.

As the screens were moved out from contact with the nozzle and placed successively farther downstream, the acoustic output at low and medium frequencies decreased at first and then remained nearly constant for $x/d = 1/2$ and larger. This behavior is partially illustrated in Figures 34 and 35 where the complete spectral curves for $x/d = 0$ and $x/d = 1/2$ are displayed. Only portions of the spectra for other values of x/d have been included to avoid a jumble of lines.

Figures 34 and 35 show that the high-frequency part of the spectrum behaves differently. Generally the amount of high-frequency noise increased with x/d. The experiments do not clearly differentiate between:—

1. a simple increase in sound power magnitude in each high-frequency band as x/d increases, and,

2. a displacement toward lower frequencies as x/d increases of a sound power maximum lying above our observable frequency range.

From Figure 35, one can find indication that both possible effects may be involved. These spectra appear to rise too steeply at high frequencies to accept the explanation that this noise is due to simple jet behavior for a nozzle diameter approximating the size of the screen's openings. (See Figure 2 for the expected shape of such a spectrum. Of course, in this case we have a laterally extended two-dimensional array of tiny jets and conceivably they might produce a steeper spectrum than the same number of completely independent sources. (However, the experiments with double nozzles reported in Section 4.3 gave no indication that such would be the case.)

On the other hand, the rate of level increase with frequency is too gradual to suggest the occurance of a single pure tone as, for example, from a screen resonance or an aeolian tone from the wires of the screen. In the present case in which the wire diameter is 0.013 inch, the aeolian tone would be expected to lie at 100 kc or higher, if the known behavior for single wires may be applied to screens. The downward shift of the high-frequency behavior with increasing downstream placement of the screen is reminiscent of a Strouhal number relationship (see Equation 3) but the appropriate velocity and linear dimension have not been identified.

A 10 mesh screen yielded results very similar to those for the 20 mesh screen as detailed above. Separate graphs will not be presented here but the data are available in Appendix III. Screens of finer mesh also gave similar results generally, although the changes in spectrum with the distance parameter were not always as distinct as those shown in Figures 34 and 35.

Experiments with a 50-mesh screen were carried to even larger values of x/d than for the 20 mesh screen and the data evidence a continuing shift of the initially high-frequency behavior toward lower frequencies; see Figures 36 through 39. As x/d exceeds two, the magnitude of the radiated sound power starts to increase at even the lowest frequencies while the spectral peak diminishes somewhat in level. Evidence of pure tone generation in the vicinity of 12,500 cps occurs at x/d = 2 and 4 for 100 SCFM (see Figures 36 and 37) and at x/d = 1 and 2 for 50 SCFM (see Figures 38 and 39). If we assume that the same free-stream velocity in the jet will occur twice as far from the nozzle for the 100 SCFM condition as for 50 SCFM, then these observations suggest that the tonal generation is limited to a specific range of flow velocity. The stability of the observed frequency with changing distance leads one to suspect a specific resonance of the screen but such has not been definitely established. This particular 50 mesh screen appears to have been more loosely stretched than most of the other screens; an observation devoid of acoustical interpretation at present.

With the exceptions already discussed, the spectra from screens appear to be remarkably free from evidence of strong resonances. For the most part, the spectra progress smoothly and gradually from one band to the next. Occasionally one band or another will fall a db or two high suggesting vestiges of resonances but resonances which are too nearly obscured by broad-band noise to have much acoustical significance. Of course, these weak acoustical effects might still be related, for example, to structural fatigue and therefore possess some indirect relevance to the engineering design of runup silencers. In as much as most of the screens were tightly stretched on mounting rings and produced tonal sounds at frequencies of a few hundred cycles per second when tapped with the finger, it is somewhat surprising that no obviously related strong noise peaks were observed in the corresponding flow-noise spectra.

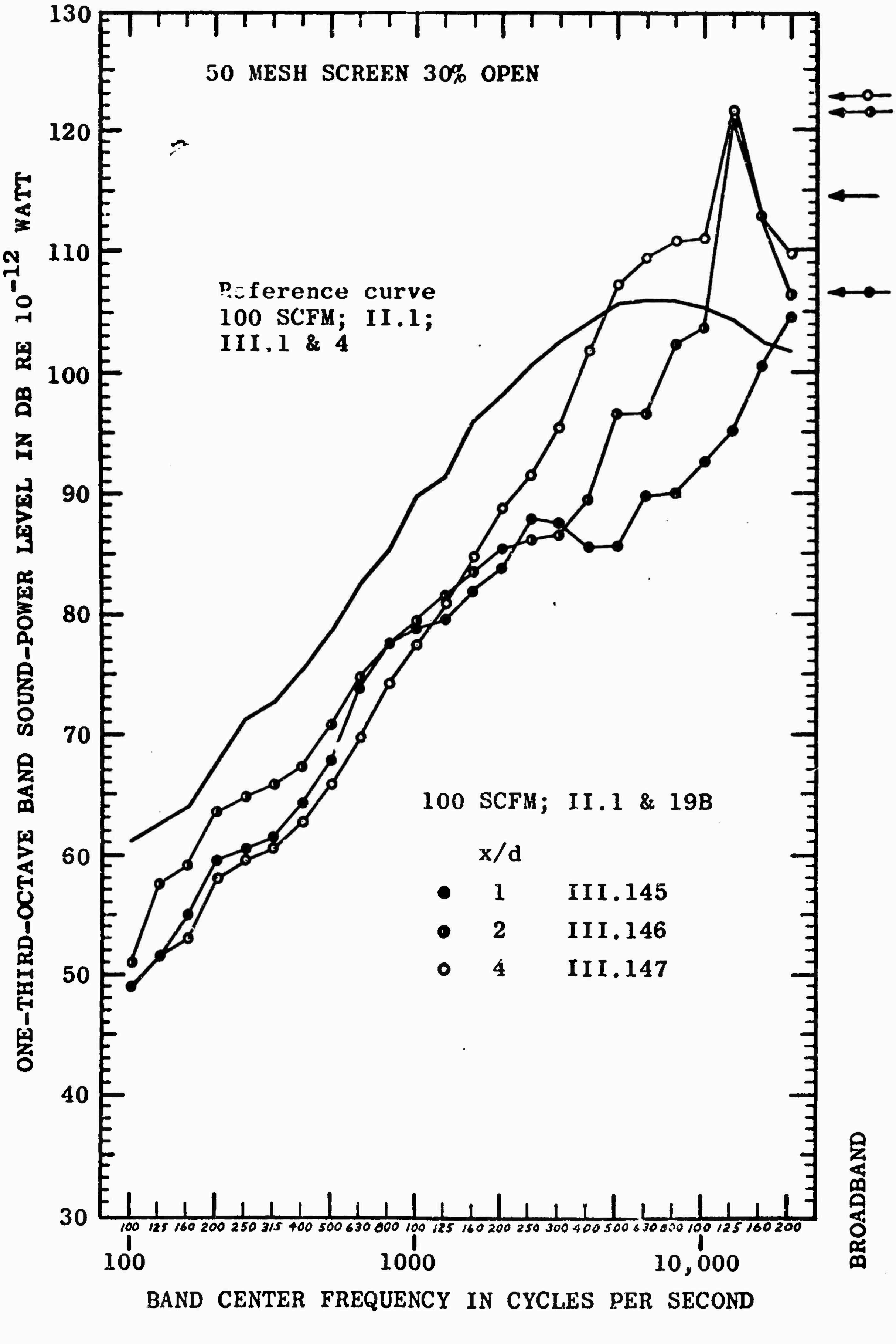

FIGURE 36. EFFECT OF DISTANCE PARAMETER; HIGH VELOCITY.

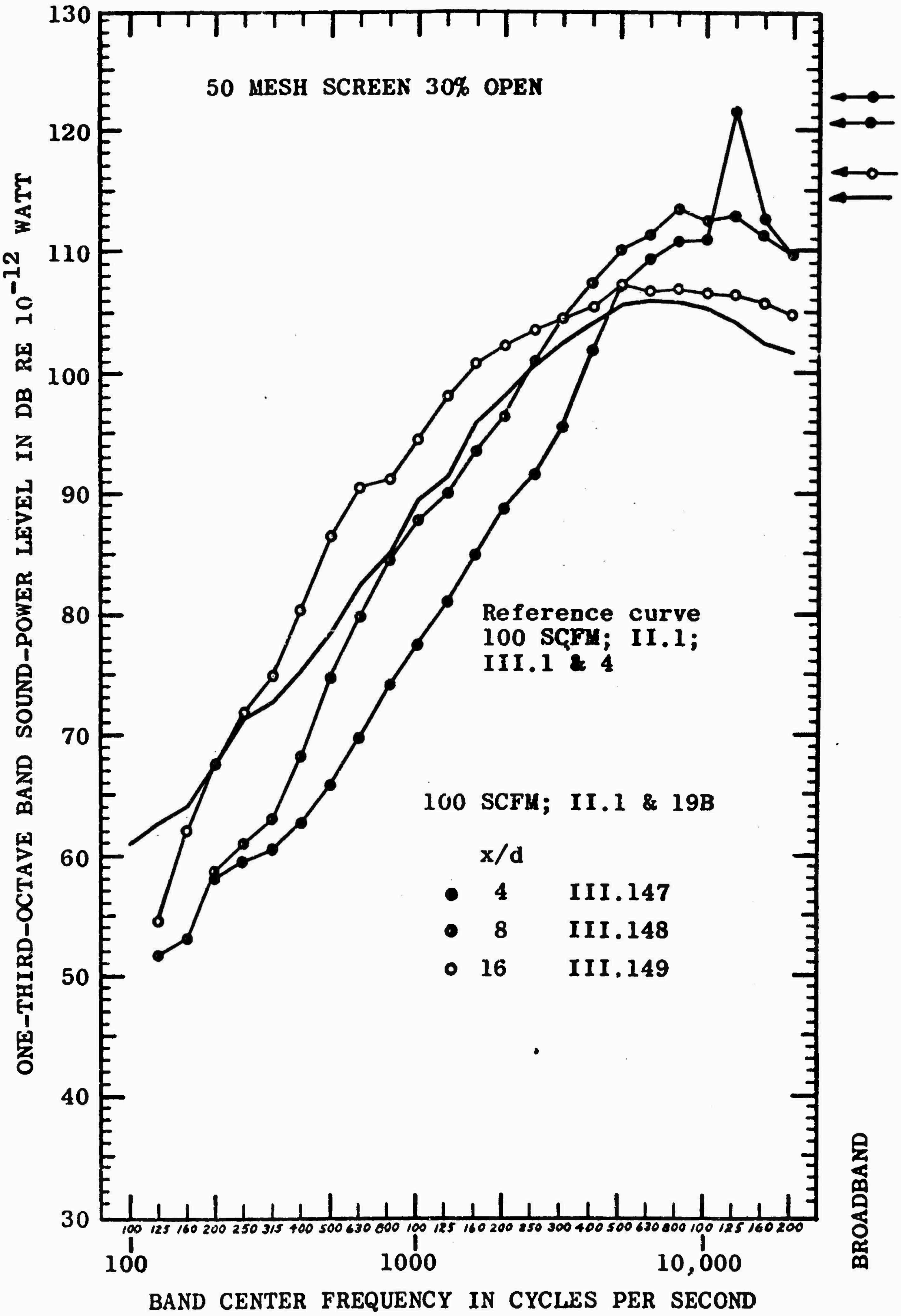

FIGURE 37. EFFECT OF DISTANCE PARAMETER; HIGH VELOCITY.

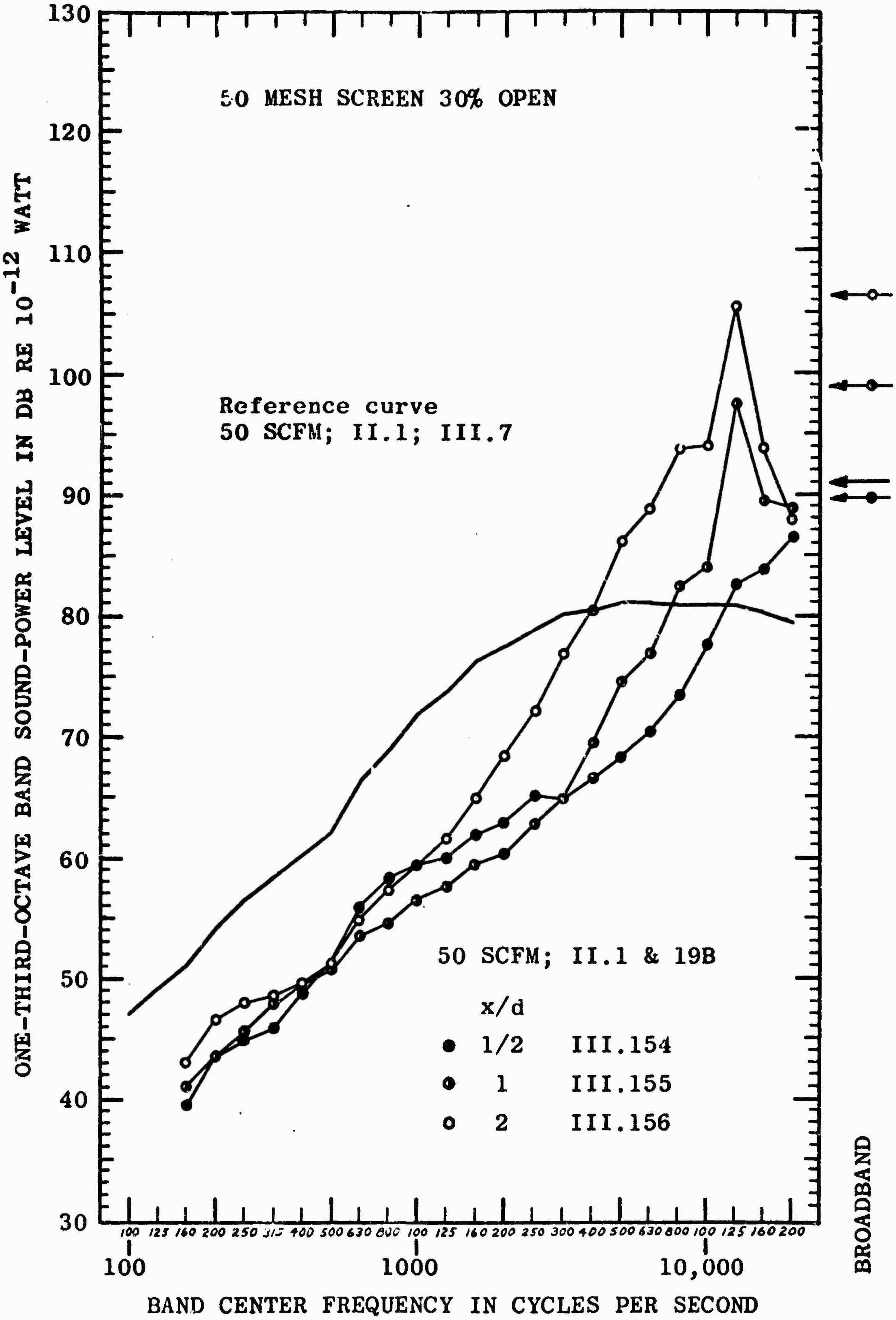

FIGURE 38. EFFECT OF DISTANCE PARAMETER; MEDIUM VELOCITY.

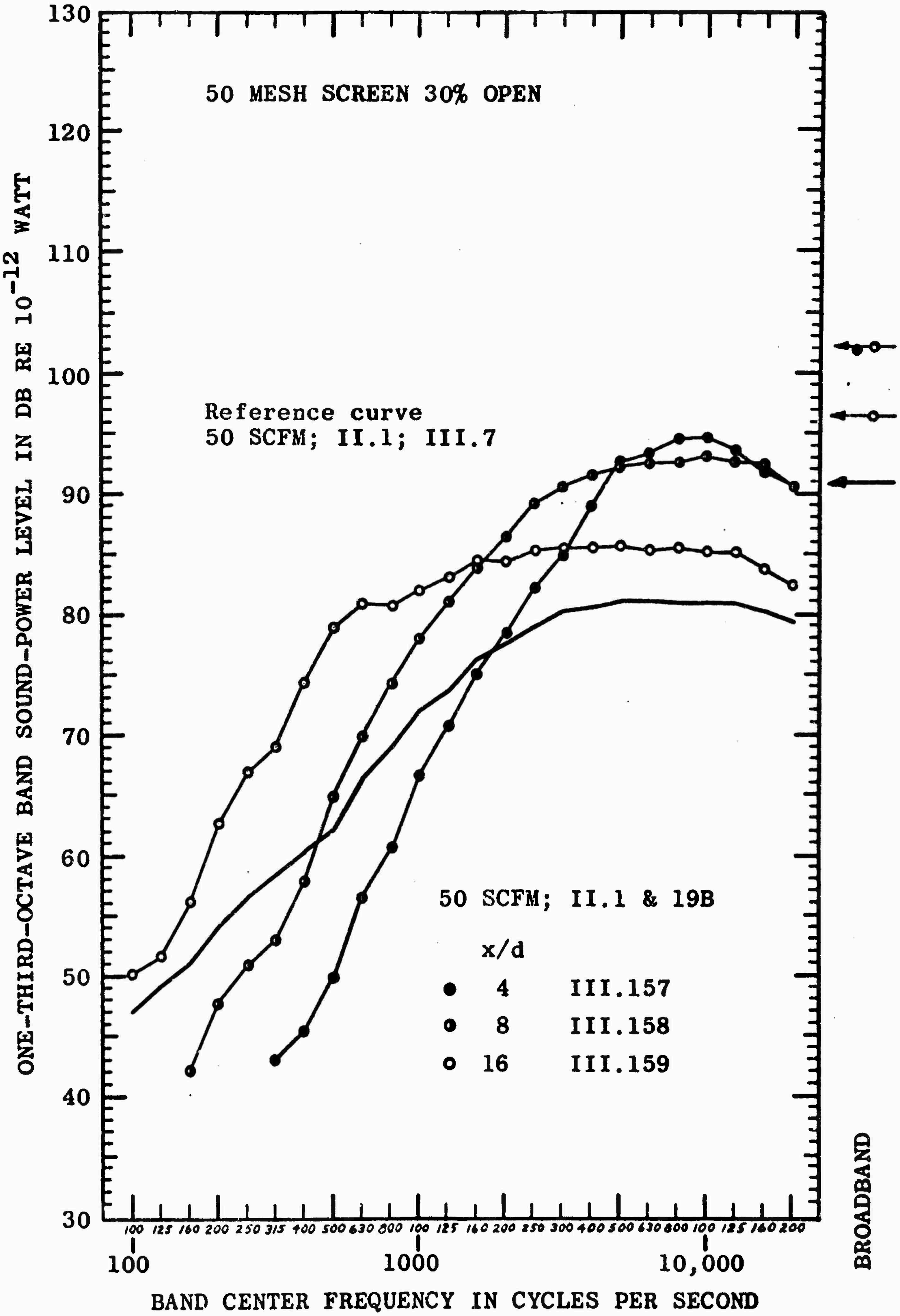

FIGURE 39. EFFECT OF DISTANCE PARAMETER; MEDIUM VELOCITY.

Figure 39 plainly demonstrates that a screen placed far downstream in a relatively low-velocity flow can increase the radiated sound power above that for the simple nozzle alone. It seems difficult also to reconcile these experimental spectra with proposed explanations that the noise should be a superposition of the noise characteristic of a simple, relatively large, jet operating between the nozzle exit and the screen, and the noise characteristic of simple jet behavior for the screen's openings at the appropriate jet velocity for the remainder of the region downstream of the screen. A somewhat different and probably more involved explanation would seem to be necessary to account for even the most prominent features of the observations reported here.

Figures 40 and 41 demonstrate the changes with distance which occurred for a 400 mesh screen. The principal features remain very similar to those already described for screens of coarser mesh. Thus, in broad outline, the acoustical effects of varying distance (here expressed in terms of x/d) as a parameter are those already stated while mesh size and percentage open area display a comparatively minor role.

Up to this point, the acoustic consequences of varying the distance separating the screens from the nozzle have been investigated using the one-half inch diameter smooth approach nozzle, nozzle II.1. Comparable investigations have been conducted using the 0.707 inch diameter and the 1.00 inch diameter nozzles (II.2 and II.3) which, of course, were operated at lower values of mean efflux velocity. Although the spectra evidence many variations among the fine details, the general trends with distance remained similar to those already reported (see III.237 through III.304). The most silencing occurred for the screens located relatively close to the nozzles and when the screens were displaced sufficiently far downstream, the amount of noise increased. A screen-to-nozzle distance corresponding to minimum noise was not as clearly delineated at these lower velocities but $x/d = 1/2$ seems to represent a reasonable choice. For the coarser mesh screens, the minimum noise may result at even somewhat smaller distances. In the opposite direction, almost without exception, $x/d \geq 1$ resulted in increased noise.

Is the ratio, x/d, an appropriate parameter for the comparison of the acoustical effects from screens when nozzles of various diameters are tested? Consideration of the gamut of acoustical data for screens indicates that this ratio is a relatively good choice provided that the comparisons are made for similar mean velocities of the unsilenced jet and for screens of the same mesh. Comparisons accomplished in this manner are not perfect in all details but certainly justify using x/d as a first approximation to a distance-like parameter.

By concentrating on a particular value of x/d, say $x/d = 1/2$, it becomes possible to examine other aspects of these screen experiments more critically. Figures 42 through 44 compare the silencing (band power levels

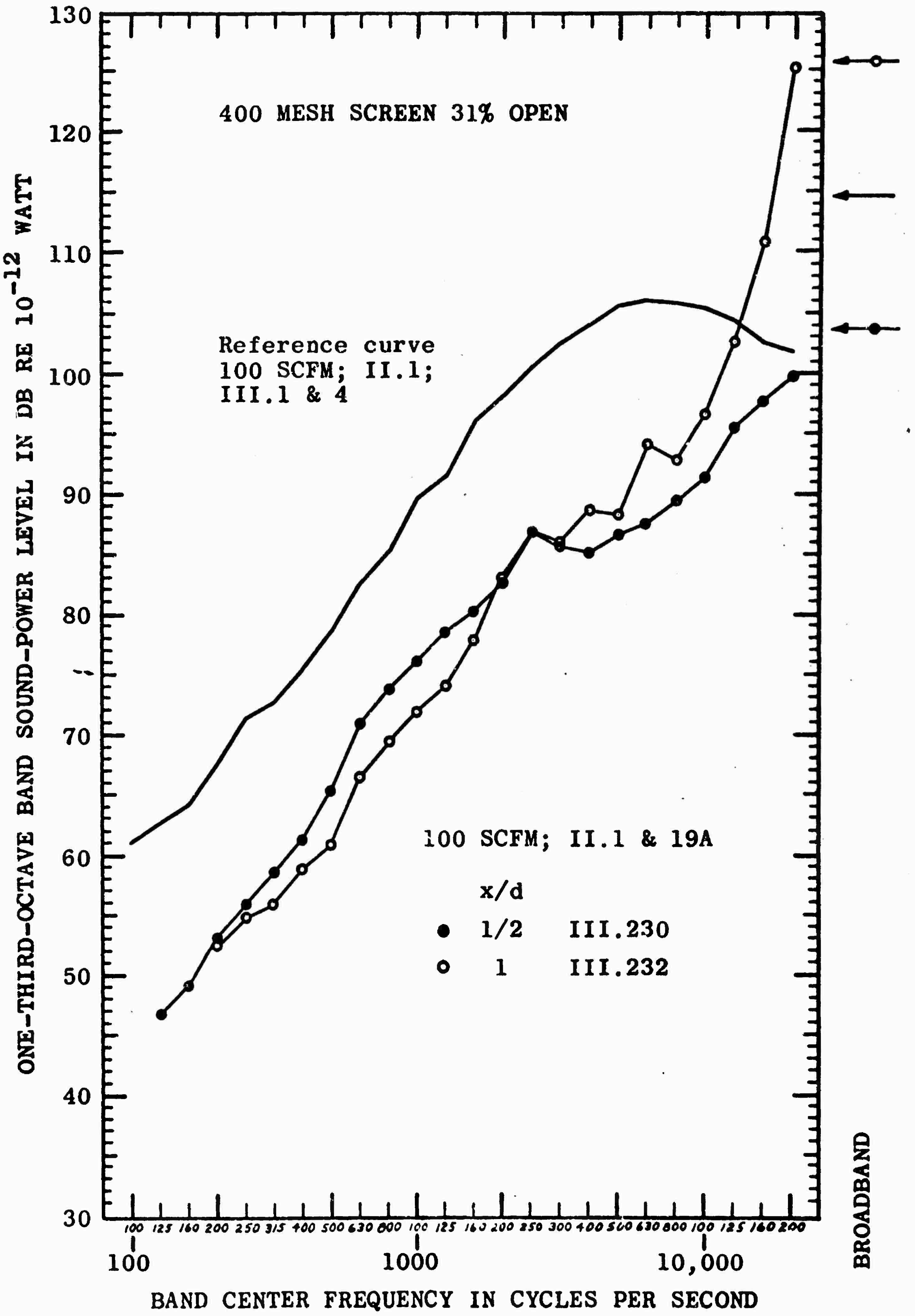

FIGURE 40. EFFECT OF DISTANCE PARAMETER; HIGH VELOCITY.

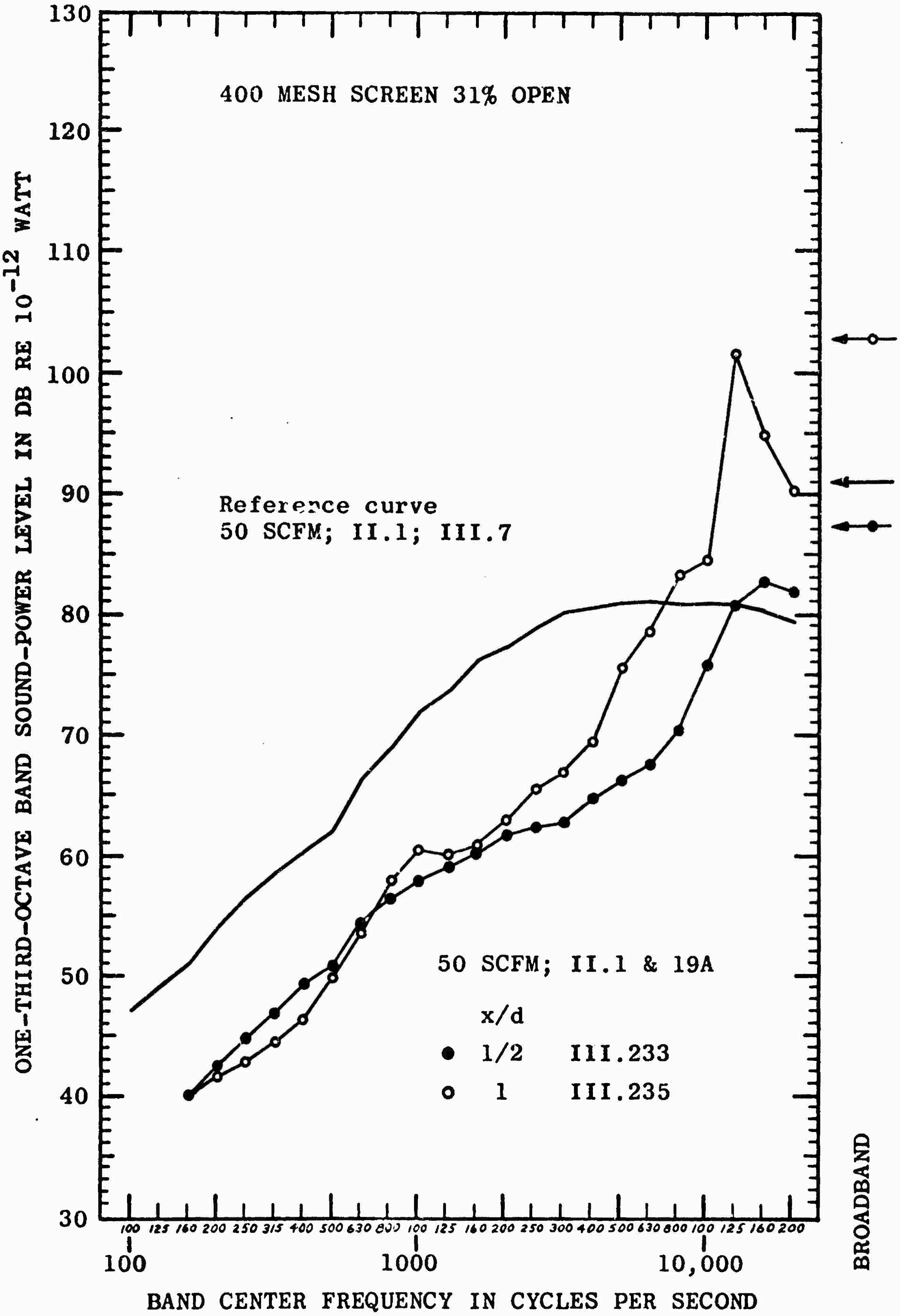

FIGURE 41. EFFECT OF DISTANCE PARAMETER; MEDIUM VELOCITY.

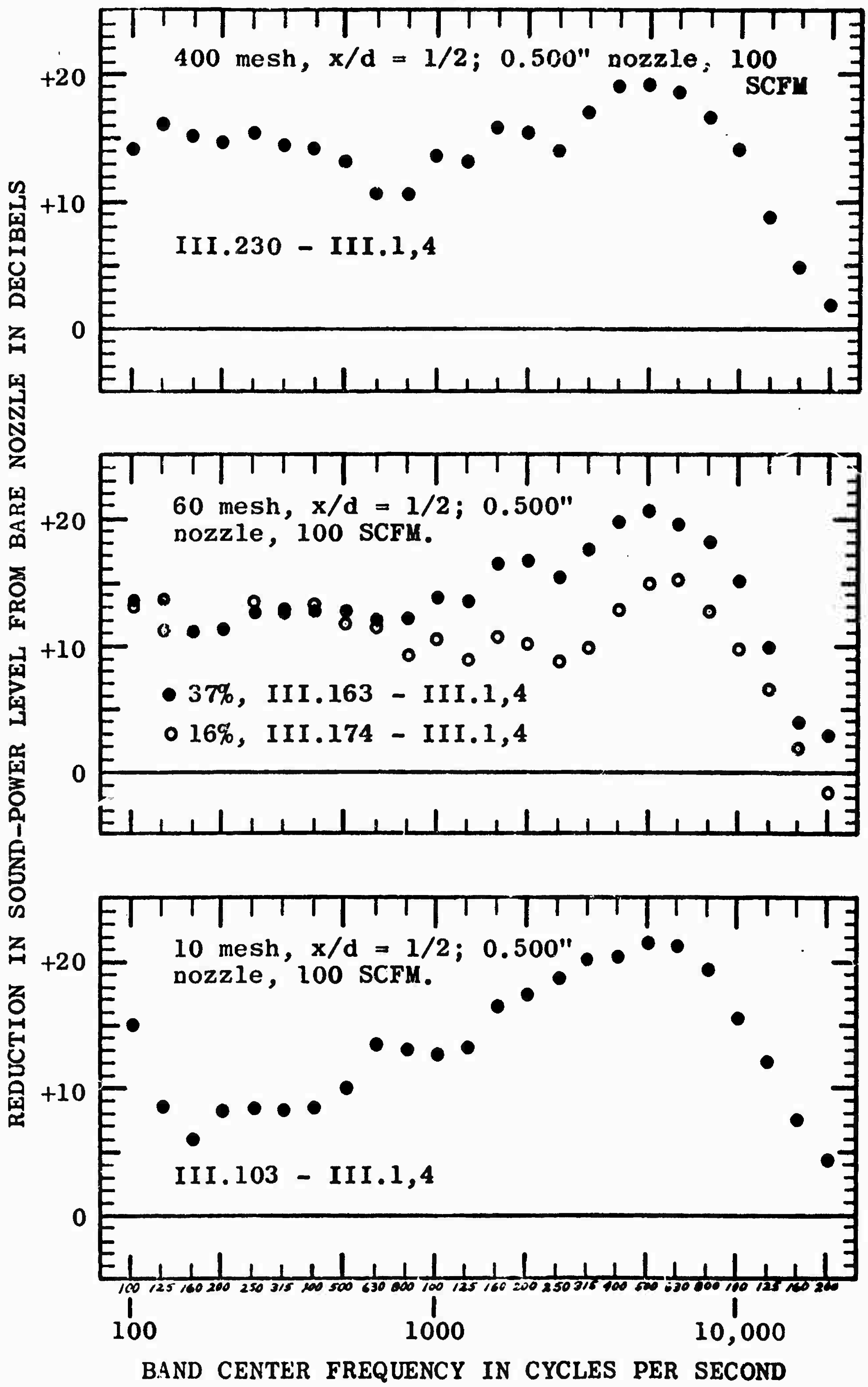

FIGURE 42. EFFECT OF MESH SIZE AT HIGH VELOCITY.

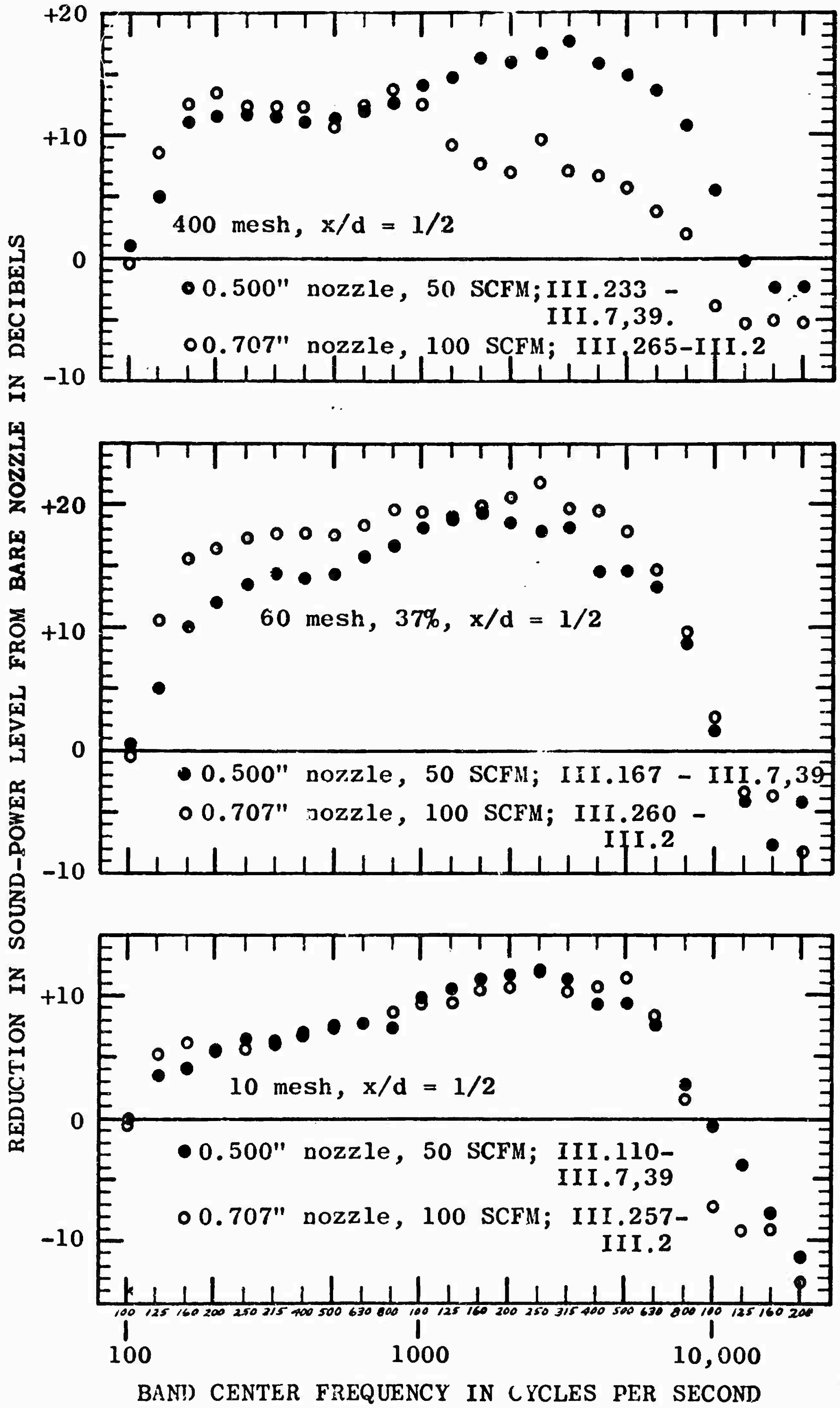

FIGURE 43. EFFECT OF MESH SIZE AT MEDIUM VELOCITY.

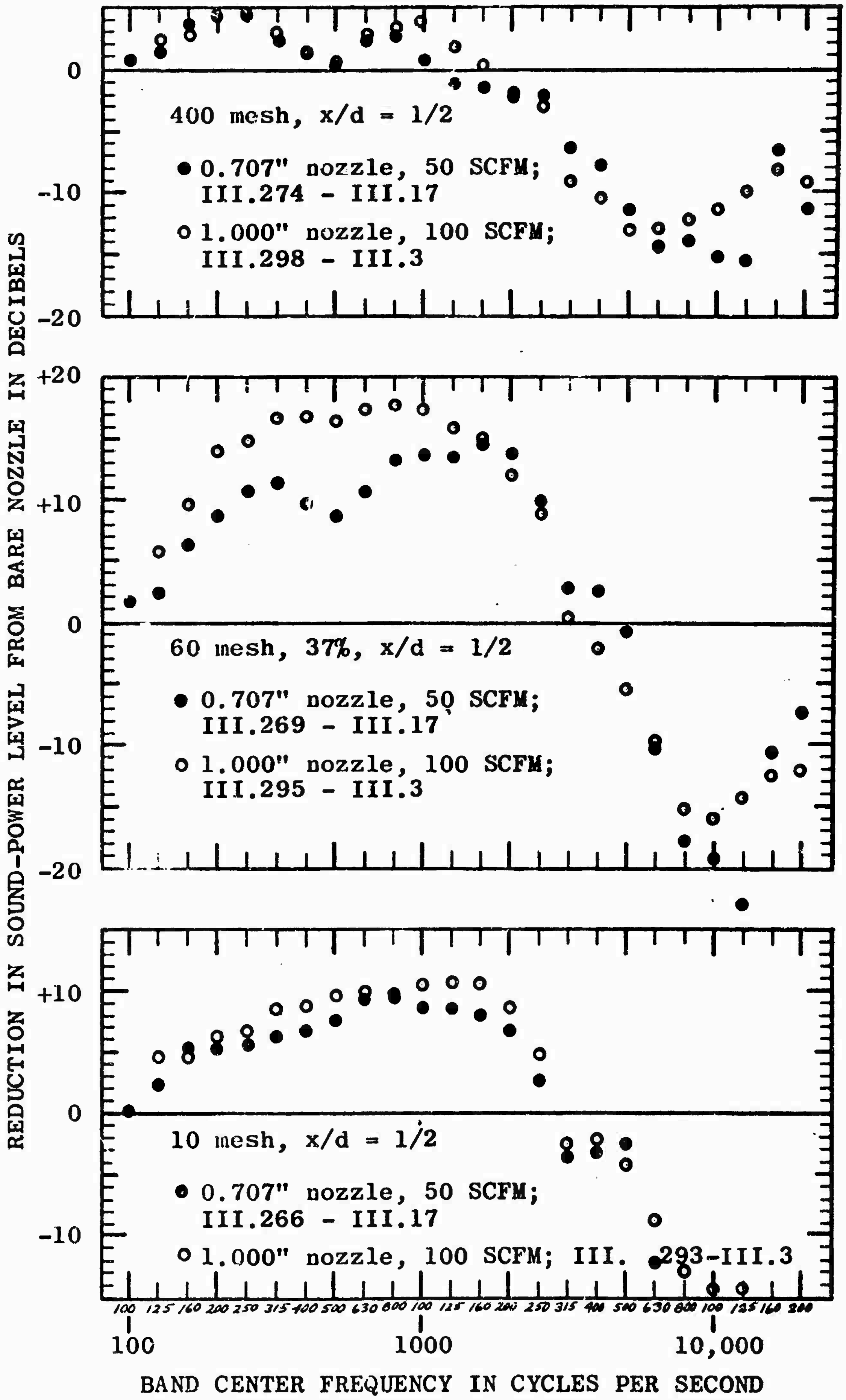

FIGURE 44. EFFECT OF MESH SIZE AT LOW VELOCITY.

radiated by the bare nozzle less the band power levels radiated with a screen in position) at three values of jet velocity, roughly Mach 1.0, 0.5 and 0.25, for a range of mesh sizes. Figures 43 and 44 also illustrate the type of results which justify using x/d as a parameter. Figure 42, which is indicative of the highest velocity experiments, displays the fact that mesh size plays a relatively minor role. A fine mesh improves the behavior at low frequencies while reducing slightly the high frequency silencing. Independently of mesh size and although still positive in sign, the magnitude of the silencing diminishes at approximately 12 db per octave at the high frequency end of the spectrum.

Figure 43 illustrates the silencing obtained for initial values of jet velocity in the vicinity of Mach 0.5. The coarsest screen, 10 mesh, is clearly seen to be too coarse for optimum silencing but the finer screens yield roughly similar results. Another point of interest is that the high-frequency fall-off has shifted downward by roughly an octave for all mesh sizes.

The downward shift of the high-frequency fall-off is even more clearly evident in Figure 44. Within the framework of these experiments, it is velocity dependent and this velocity dependence is over and above that ascribable to simple jet noise behavior. Figure 44 demonstrates a clear superiority for the 60-mesh screen at a relatively low velocity, both coarser and finer screens producing less silencing.

Figures 43 and 44 appear to show that the silencing due to screens drops to essentially zero at the lowest frequencies whereas in Figure 42, this did not occur. In reality, the drop-off at low frequencies results from background noise limitations which afflict the relatively quiet configurations most. This trouble constitutes one of the worst disadvantages of the graphical format used in Figures 42 through 44 compared to the ordinary sound-power-spectrum format used in many of the other figures. The experiments are not definitive with respect to the very low frequency behavior of screens but indications are that some silencing probably continues to occur.

The matter of the wire diameter for a screen of given mesh, that is, the percentage of open area, is illustrated in Figure 45 for two 60 mesh screens operated at several values of jet velocity. The more open screen (37% open) is superior to the more closed screen (16% open). Table 3 would cause one to expect that a similar direct comparison could be made for the 25% and 33% open 200 mesh screens. However, the differences observed for these two screens were to small to constitute a definitive acoustical result. This result is probably a consequence of the relatively small difference in percentage open area as well as of the fact that 200 mesh is quite far removed from optimum mesh size for some of the experimental situations. The total evidence suggests that the more open screens provide more silencing.

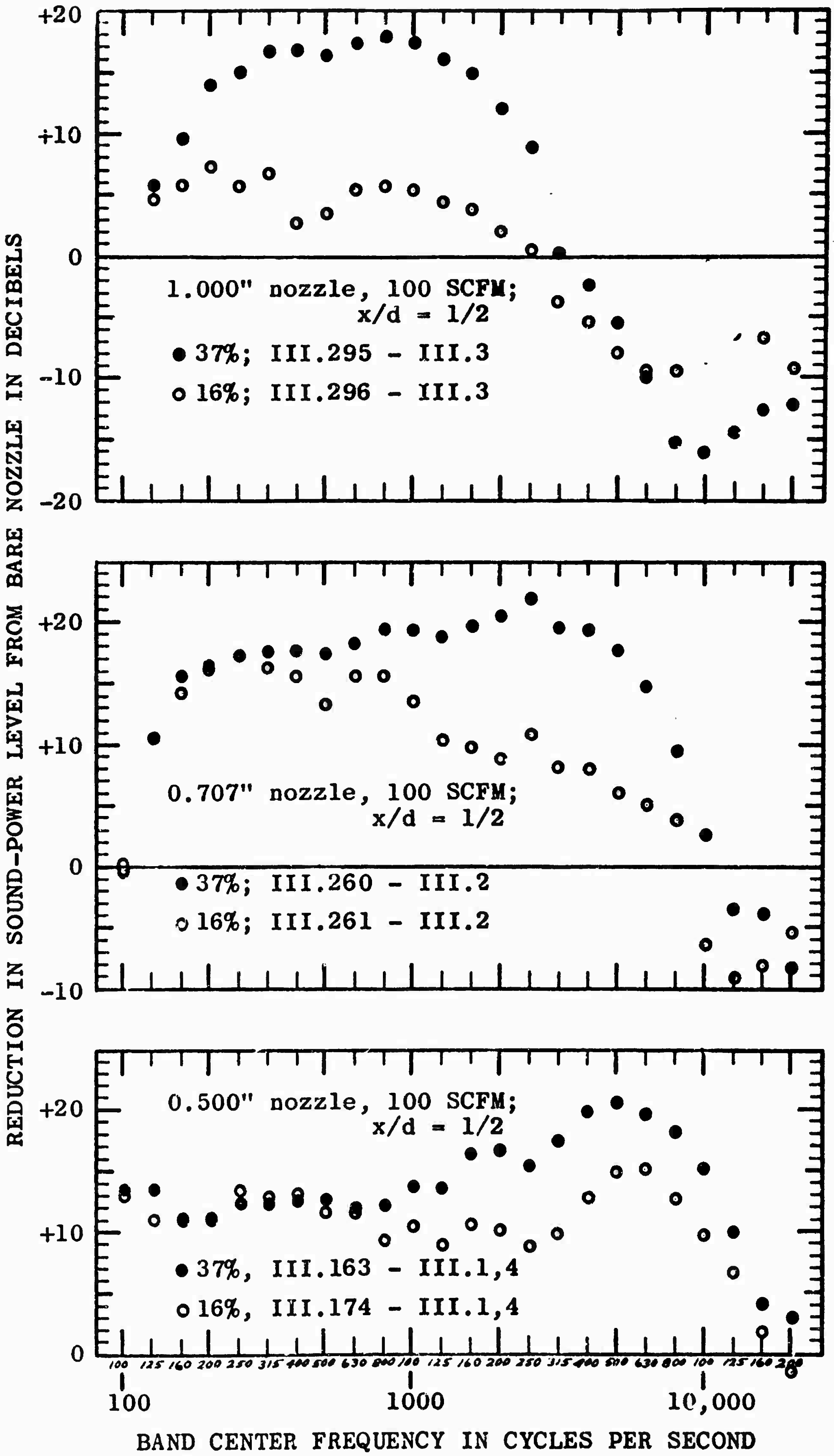

FIGURE 45. EFFECT OF PERCENTAGE OPEN AREA; 60 MESH SCREENS.

In contrast to mesh size, the percentage open area represents a variable which could not be readily selected in the desired increments from the normal stocks of wire-cloth manufacturers' and Table 3 reflects this situation. Many of the available screens just happen to be about 30% open, particularly from 50 mesh upward. The coarser screens tended to be more open and so the acoustical results for them are compounded of the effects of changes both in mesh size and open area. In future experiments, it may be advisable to have some of the screens specially fabricated so that mesh size and percent of open area can be incremented by steps of specific and comparable magnitude. For instance, within the size range of the present research, one might concentrate on meshes between 10 and 100 and percentage of open area from say 25% to perhaps 75%.

Figure 46 demonstrates the acoustic results of placing a screen in the nozzle exit for subsonic flow conditions. It is evident that very little acoustical change was produced except for a slight increase in the high-frequency spectrum; to a first approximation the screen had no acoustical effect. Without the screen, 100 SCFM through the 1.00 inch diameter smooth approach nozzle would produce approximately a Mach 0.25 mean flow velocity. With the screen in place, and considering the percentage of open area, the individual little jets might be expected to have a mean velocity of Mach 0.83 whereas the observed pressure ratio across the nozzle indicates roughly Mach 0.70. (This finding is consistent with the result noted previously that consideration of the projected area and the observed pressure ratio did not fully agree.) In the present situation, it seems as if the many small high-velocity jets created by the screen must have coalesced into a single large low-velocity jet before much acoustic radiation could occur.

Several experiments were aimed at determining the minimum lateral extent of a screen without compromising its acoustical effectiveness. One method of investigation was to cover the screen with a restrictive aperture and then observe the acoustical consequences. This method is not completely satisfactory because the resulting geometrical configurations can introduce extraneous acoustical consequences. That is, the face of the nozzle, the spacer ring for the screen, and the plate containing the restrictive aperture constitute an acoustical cavity. Nevertheless, the experiments of this type were moderately successful when interpretation was restricted to general trends. Figure 47 shows the results for a 1/2-inch diameter nozzle operated at 100 SCFM and with a 30-mesh screen located at $x/d = 1/2$. The upper solid line represents the spectrum for the bare nozzle while the lower line is for the screen having essentially unrestricted lateral extent. The several types of circular dots correspond to cover plates with different diameters of restrictive circular aperature. A one-have inch aperture produces essentially the same spectrum as the bare nozzle; a result to be expected. The one inch aperture yields almost the same spectrum as the unrestricted screen except that there are vestiges of other effects probably attributable to the presence of a cavity. One may conclude in this case that the full effectiveness of the screen can be realized if the diameter of the screen is about twice that of the nozzle exit.

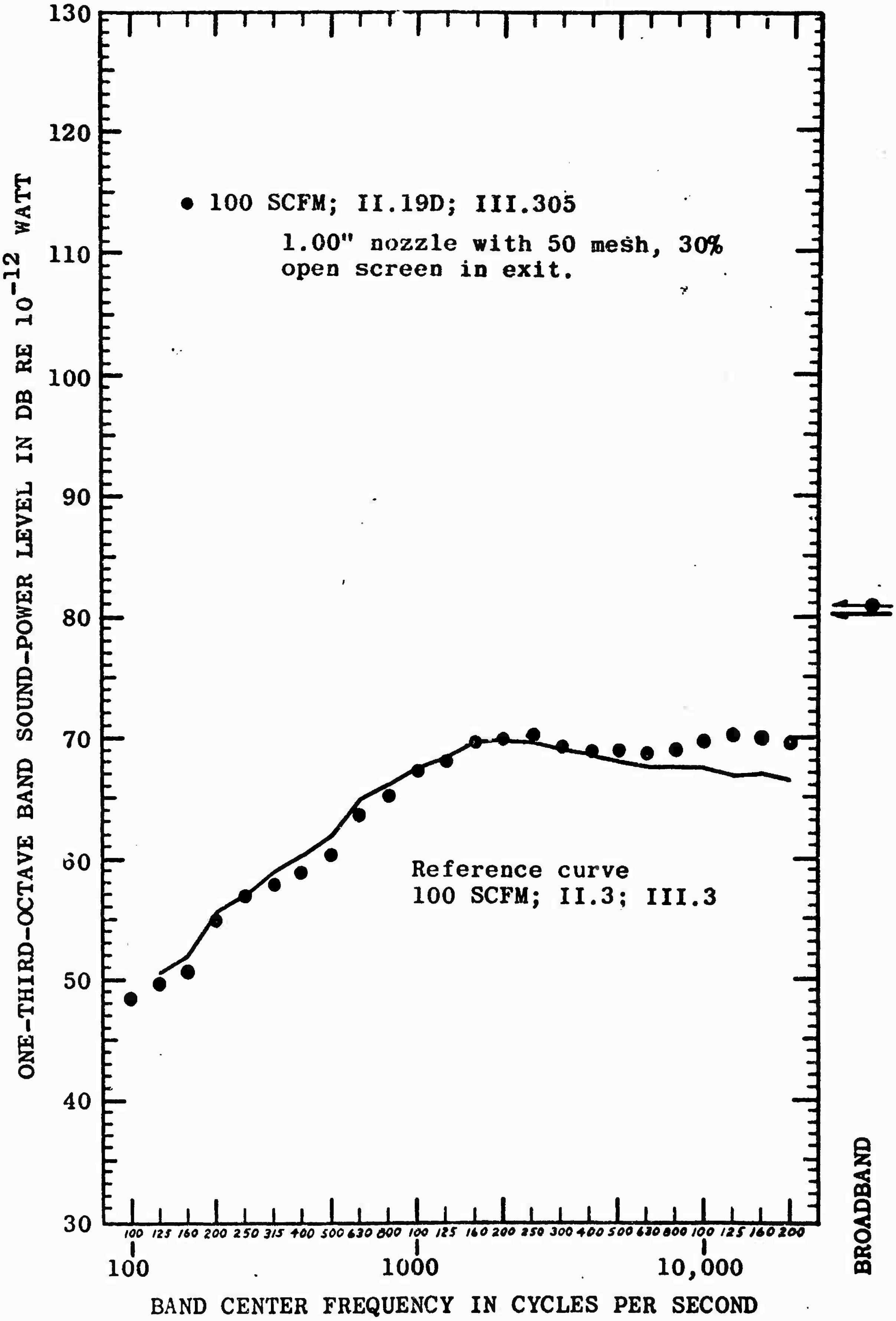

FIGURE 46. SCREEN IN NOZZLE EXIT; LOW VELOCITY.

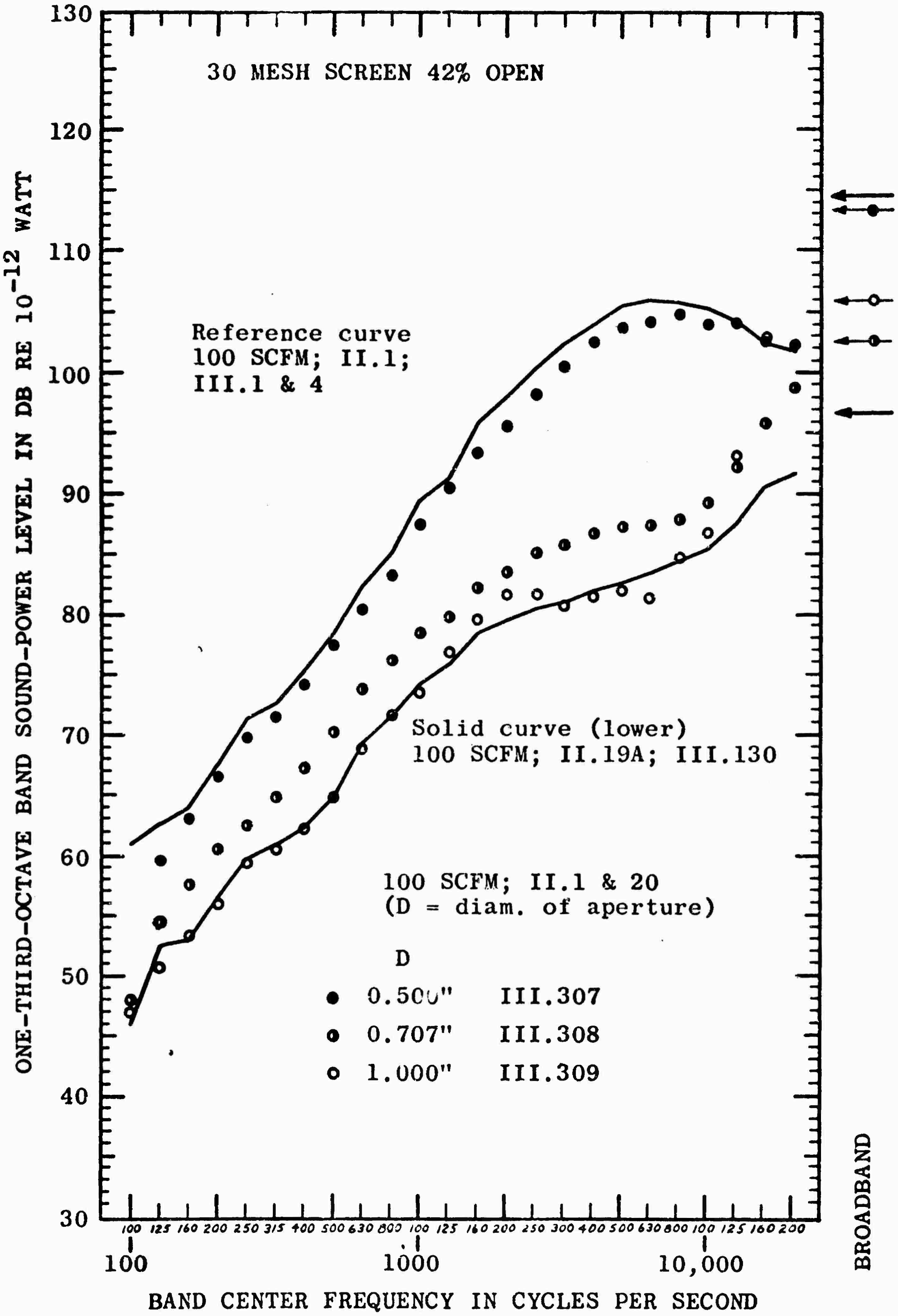

FIGURE 47. EFFECT OF COVER PLATE OVER SCREEN; HIGH VELOCITY.

Cover plates with larger apertures were tested, but except for evidence of acoustical cavity behavior, they yielded results similar to the spectrum for the unrestricted screens. Figure 48 displays similar results for 50 SCFM through the 1/2-inch diameter nozzle. In this instance, an aperture of 0.707 inch diameter appears to be only slightly limiting. Cavity effects are more predominate for the one-inch than for smaller apertures.

These experiments, considered as a whole, suggest that a screen needs to be only twice the diameter of the jet exit if the screen is located in the vicinity of $x/d = 1/2$. Screens having larger lateral dimensions do not appear to possess any acoustical advantage to warrant the larger physical structure. Of course, optimization of a full-sized design would have to be achieved by normal developmental engineering procedures but as a first estimate, one would expect physical dimensions consistent with the above test results.

Figure 49 displays the acoustical consequences of exhausting the 1/2-inch diameter model jet into a long screen cylinder fabricated of 80 mesh screen which has roughly 25% open area. Noise generation was markedly enhanced especially for the lower-velocity test condition. (This particular configuration of screen was a sample provided by a local company and represented an item fabricated for some other purpose.) This screen cylinder has some similarity to a Tyler muffler (see Reference 23) but, of course, the openings in the screen are more closely spaced.

Several configurations (screen cylinders, cones and paraboloidal shapes) of suitable size and mesh were selected and experimented with, but none of them gave any indication of useful noise reduction. There was no a priori reason for anticipating noise reduction but it was desirable to experiment with a wide range of configurations which could hardly be investigated systematically. These samples, fabricated for other applications, represented an expedient source of unusual configurations.

During some of the preliminary experiments with screens, it was observed that a screen could be placed at an angle across the jet without much change in the acoustical result although a considerable proportion of the jet flow was deflected to the side. This arrangement appeared to have the potential of being developed into a very simple exhaust deflector which would provide appreciable silencing. Accordingly several experiments were conducted in which various screens were positioned at 45° with respect to the jet axis. Because of the necessity of locating the screen close to the nozzle, it was placed in contact with one edge of the nozzle extension; thus these screens were at $x/d = 1/2$ measured on axis.

Figure 50 shows the results for a 20 mesh screen and 100 SCFM of air. The upper solid curve is the spectrum for the bare nozzle while the lower broken curve represents the test results when the 20 mesh screen was normal to the axis. The dots present the spectrum for the 20 mesh screen tilted at 45° to the jets' axis and with the short extension tube attached to the

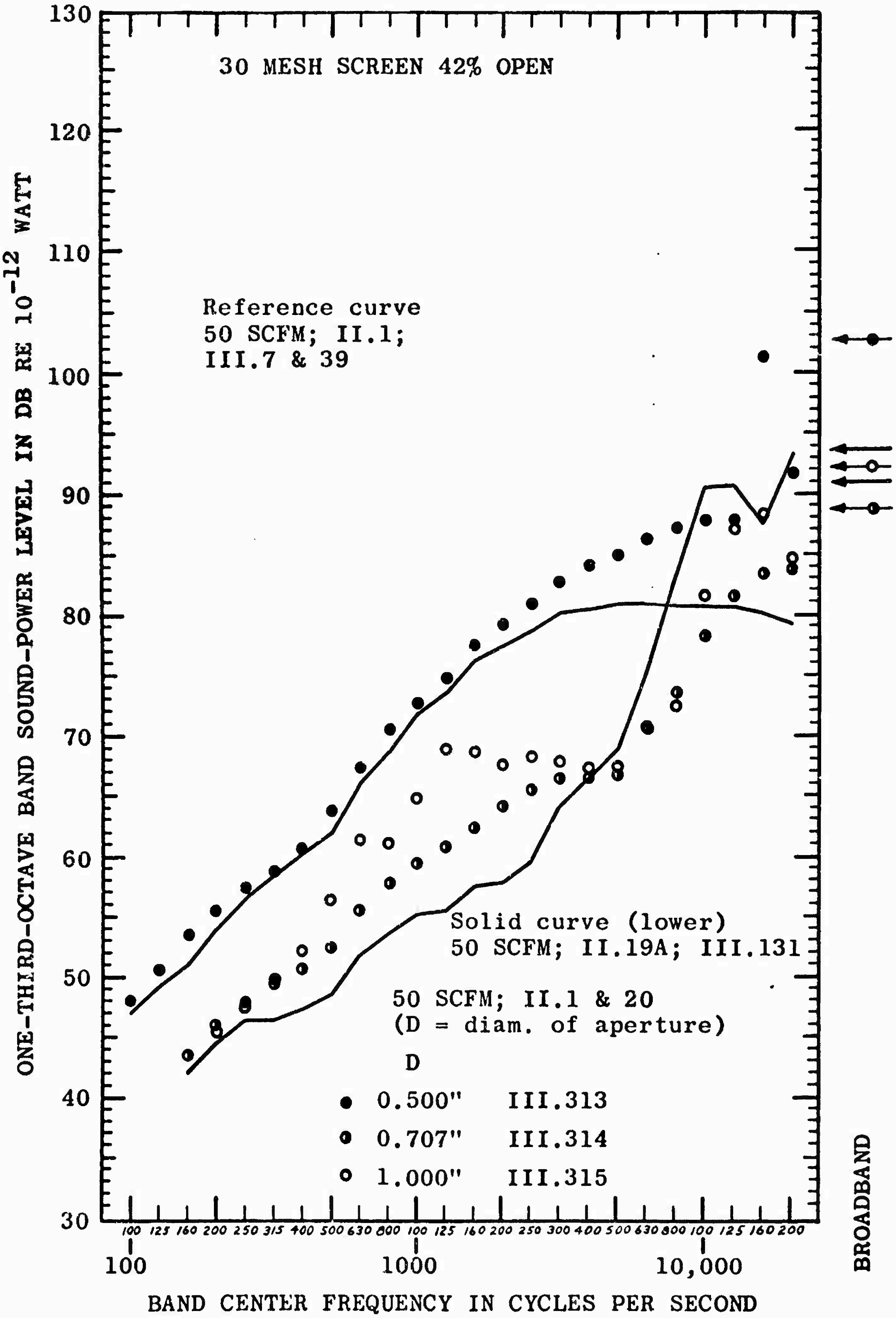

FIGURE 48. EFFECT OF COVER PLATE OVER SCREEN; MEDIUM VELOCITY.

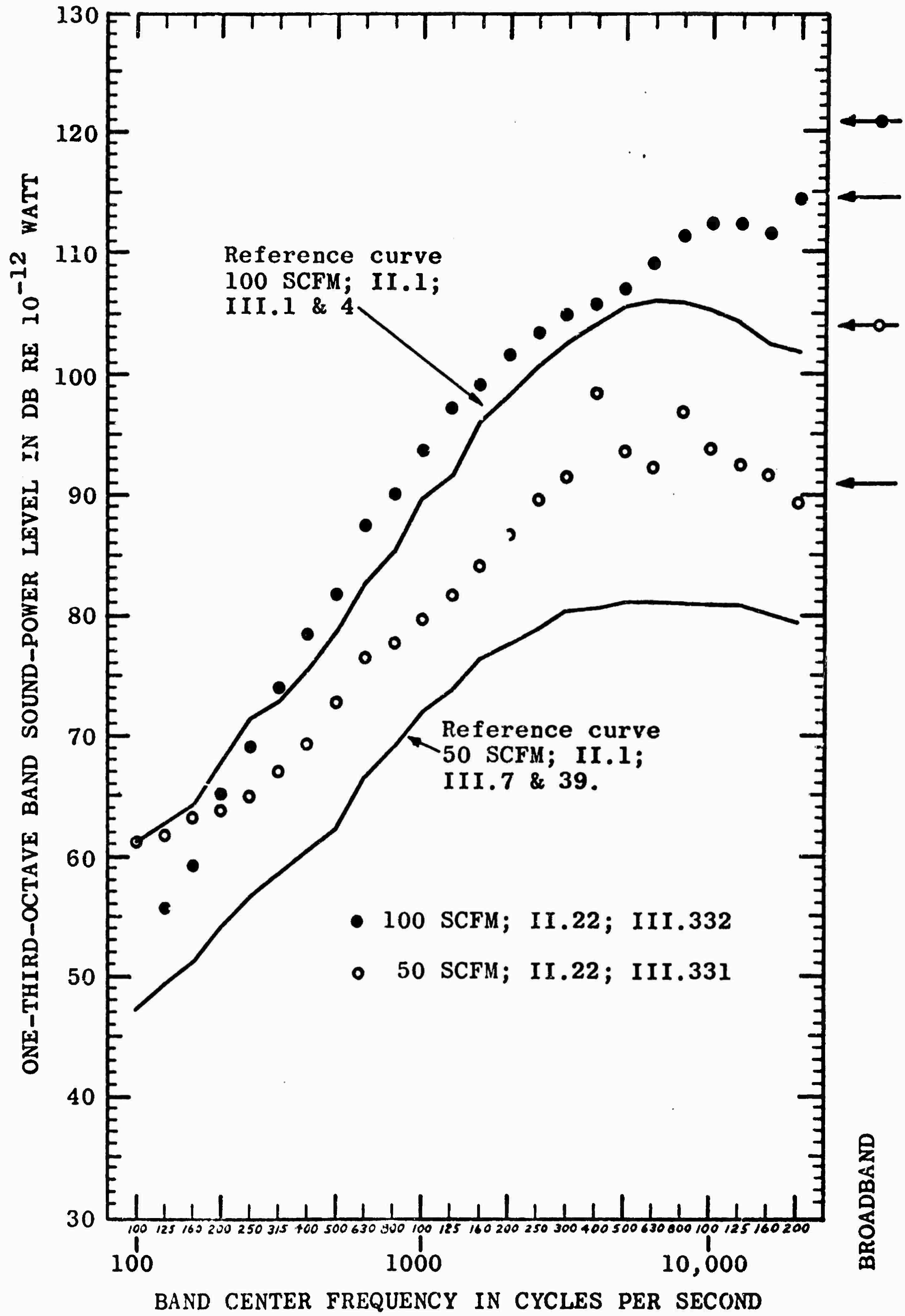

FIGURE 49. JET EXHAUSTING INTO LONG SCREEN CYLINDER.

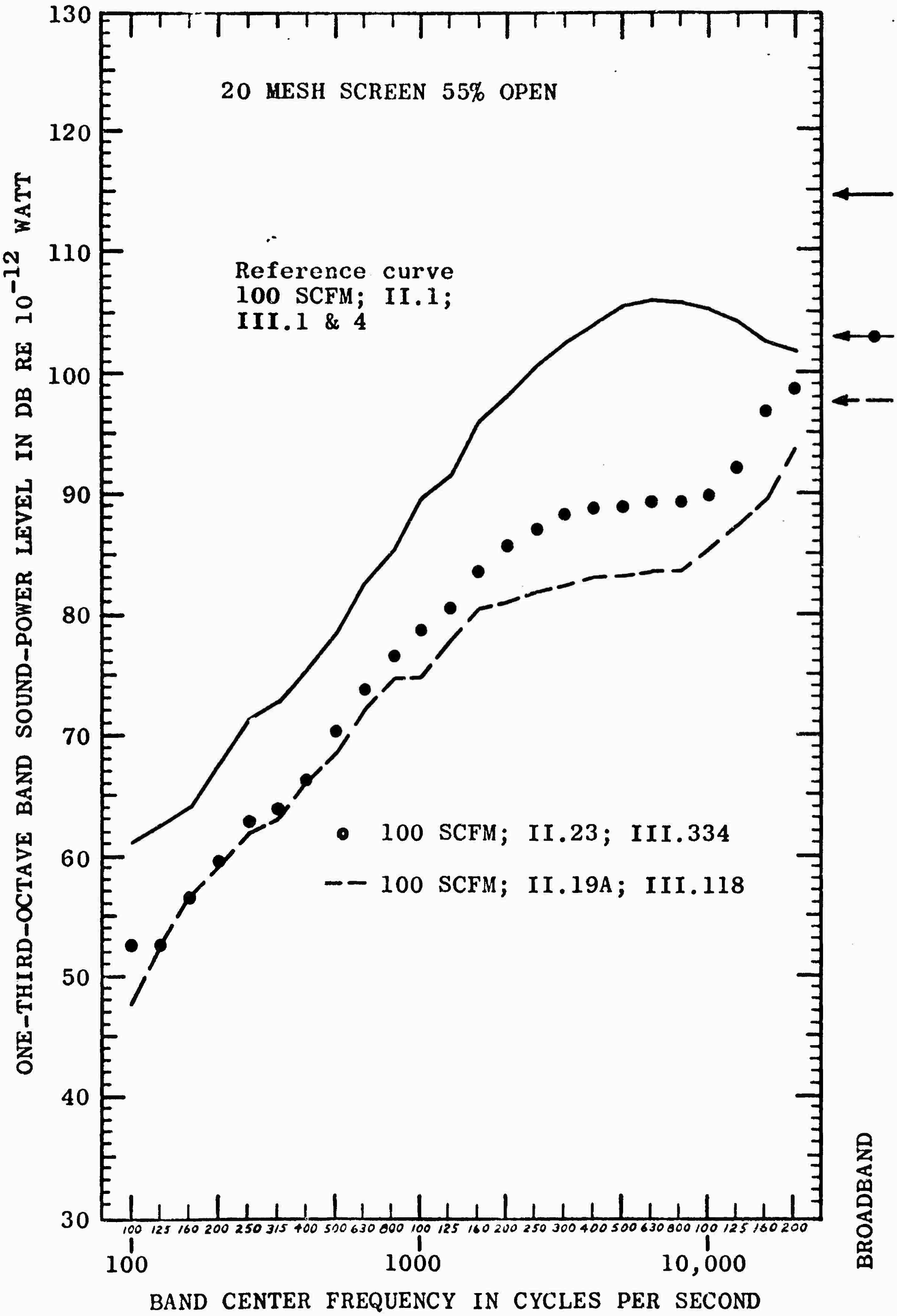

FIGURE 50. SCREEN, 20 MESH, TILTED 45°; HIGH VELOCITY.

nozzle. No correction has been made to the spectrum for the noise generated by the short extension tube because, as demonstrated in Figure 24, the difference is small at 100 SCFM.

Figure 50 indicates some loss in effectiveness caused by tilting the screen but a very useful amount of silencing remains and presists down to the lowest frequencies. Indeed below 2000 cps, the silencing averages approximately 10 db; an amount which under ordinary listening conditions corresponds to halving the subjective loudness. Of course, Figure 50 also suggests a possible silencing deficiency at very high frequencies. Nevertheless, there is a very real amount of silencing demonstrated as compared to the results for a solid-plate deflector shown earlier in Figure 32.

Figure 51 presents a similar comparison of the effectiveness of a 20 mesh screen at a lower initial velocity of the jet. Again, the solid and broken lines represent the acoustic spectrum for the bare nozzle and for the screen placed normal to the jet axis respectively. The dots represent the acoustic output of the screen tilted at 45° to the jet axis but in this case, an adjustment for the noise due to the short extension tube, see Figure 24, was necessary. The difference in level between the extension tube and extension tube with screen was determined and the dots were plotted this amount below the reference curve for the bare nozzle. Therefore, the dots in Figure 51 represent adjusted data and not the directly comparable experimental results as usually plotted. Assuming that the adjustment is a valid one, then the tilted screen yields appreciable silencing over most of the frequency range and represents a very substantial improvement over a solid exhaust deflector plate, see Figure 33.

Figures 52 and 53 show the acoustical results for 60 mesh screens angled at 45° behind the 1/2-inch diameter short extension tube. The principal reason for displaying these particular results is the clear and strong dependence upon the openness of the screens; solid and open dots representing 37% and 16% open area respectively. This dependence is more clearly evidenced by the tilted screens than it was for these same screens placed normal to the axis.

The effectiveness of the 60 mesh, 37% open screen is not unlike that of the 20 mesh screen. Experiments were conducted with finer mesh screens also but these yielded less promising results. The indications are that future research with angled screens should concentrate on the coarse mesh relatively open screens.

Because the concensus of results with screens pointed in the direction of coarse, open screens, several additional experiments were conducted, starting with a 10 mesh, 56% open screen and then removing alternate wires from it. In this way, screens having 5 x 5, 5 x 10 and 0 x 10 mesh configurations were produced and they were tested at $x/d = 1/2$ and normal to the axis (see III.345 - 352). The acoustical results fell between those [illegible] the

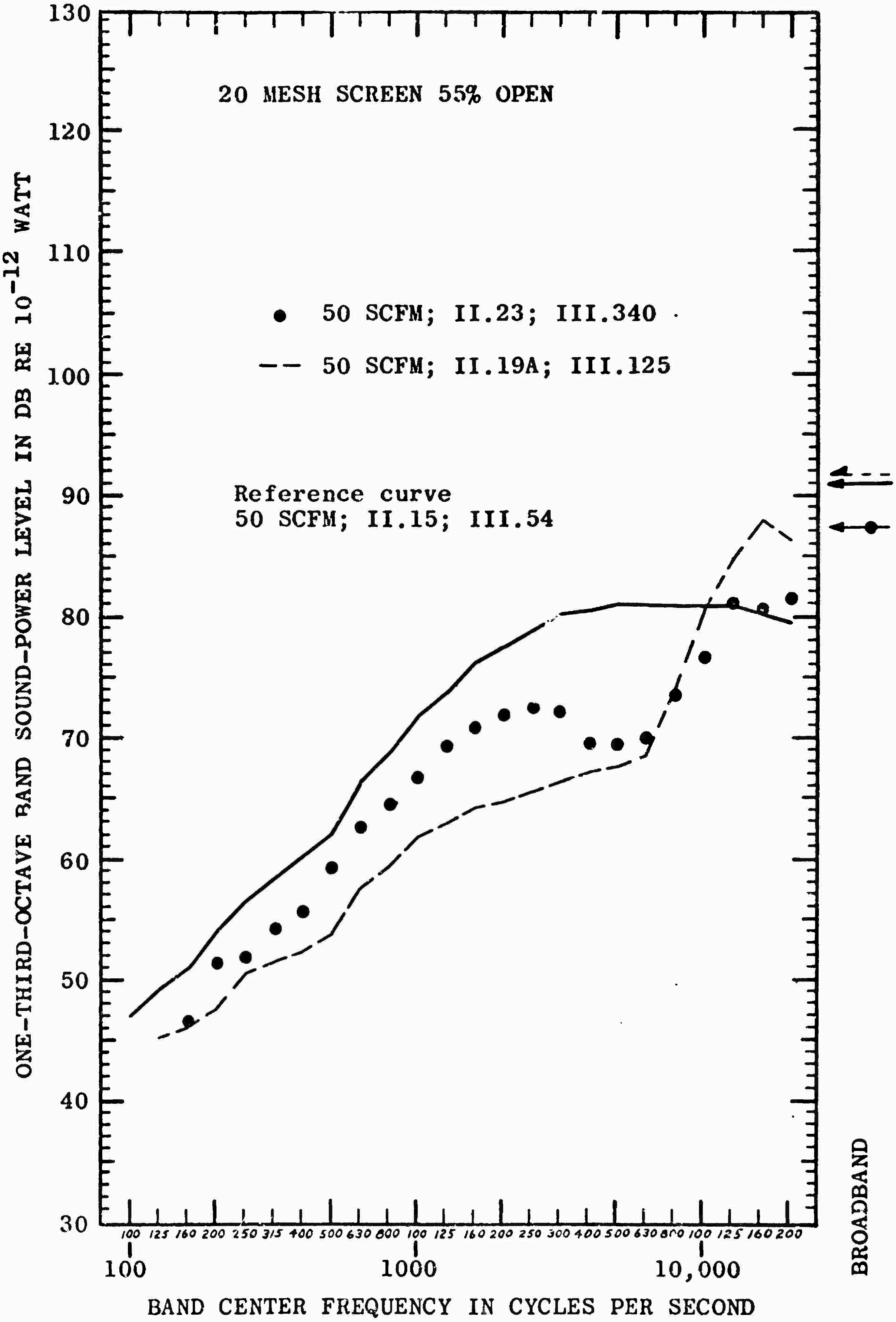

FIGURE 51. SCREEN, 20 MESH, TILTED 45°; MEDIUM VELOCITY.

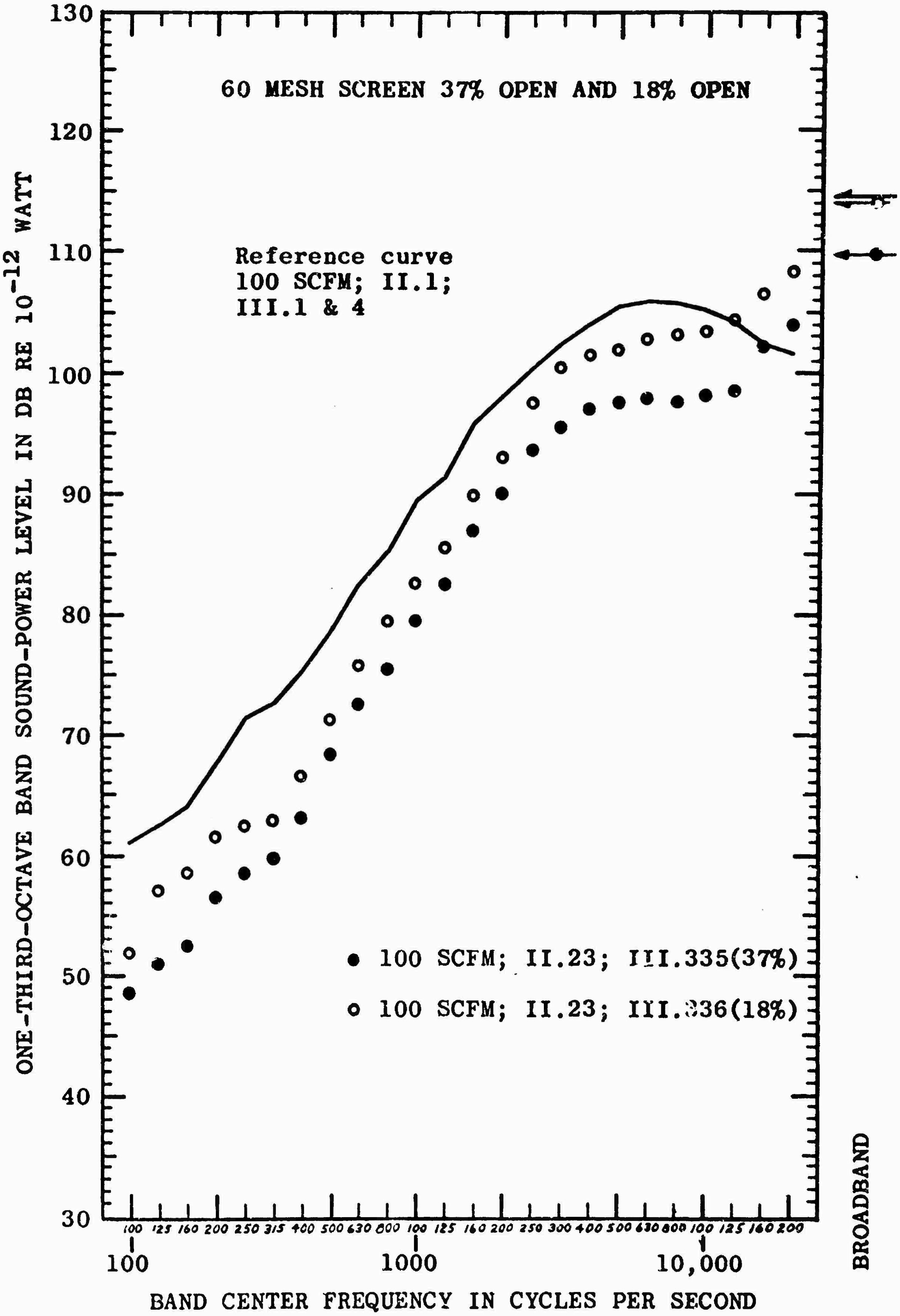

FIGURE 52. SCREENS, 60 MESH, TILTED 45°; HIGH VELOCITY.

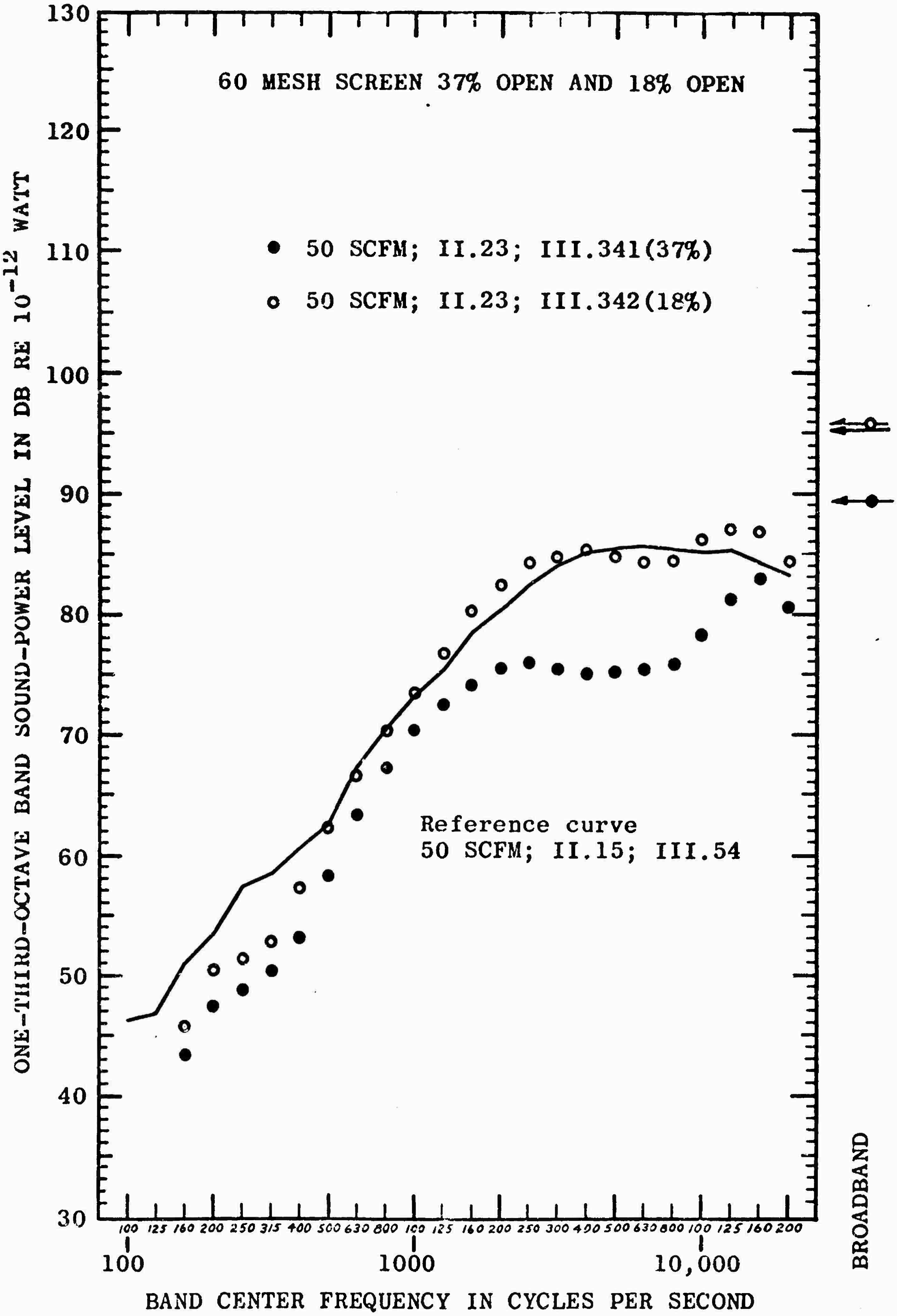

FIGURE 53. SCREENS, 60 MESH, TILTED 45°; MEDIUM VELOCITY.

bare nozzle and for the 10 x 10 mesh screen. Apparently these altered screens had passed the point of diminishing returns and there was too small an amount of wire exposed to the jet to create the maximum effect.

The method of measuring the sound power spectrum employed in this research proved to be so sensitive and reproducible that it appeared possible to detect the spectral changes due to a single rod or wire stretched across a high velocity jet. (This presumption does not refer to aeolian tone experiments in which the existance of a single pure tone is relatively easy to detect.) To test this contention, several experiments were performed with round rods, 1/16 inch and 1/8 inch diameter, placed diametrally across the jet flow at several distances from the nozzle (see III.319 - III.330). As Figure 54 confirms, the acoustical consequences of a single wire can be detected and studied. Locating the wire further downstream tended to increase the noise and the 1/8 inch diameter rod tended to be noisier than the 1/16 inch diameter rod.

Insufficient research time was available to continue this particular line of investigation but it appears to offer a definite possibility for future research. The comparative simplicity of a single wire configuration may provide a tractable point of departure for some theoretical studies. Experimentally, it would be possible to fabricate a screen wire by wire while following the acoustical consequences in detail. Similar experiments might investigate much finer single wires, single wires or bars placed other than diametrally and normal, two wires placed parallel to one another, two wires crossed (is it the total length of wire exposed to the jet or does an intersection have a beneficial effect), cross-sectional shapes other than circular, annular or radial configurations of wires, etc. Research with single wires and rods might be very informative in as much as aeronautical drag data exist for some profiles. Reference 18 speculated that bars with streamlined cross-section might yield better results than round bars while Reference 19 subsequently demonstrated that streamlined bars produced less silencing. Possibly one should try the opposite approach of maximizing the drag or maybe the circular profile is already an optimum shape. These and many other facets of the problem appear capable of systematic investigation using sound power measurement by the reverberation room method. The acoustical effect of a single wire or bar as shown in Figure 54 is rather striking in comparison to the amount obtained from a complete screen.

One final observation about screens is worth mentioning because of its apparent consequence to free-field measurements. During some of our studies on the adequacy of diffusion in the reverberation room at frequencies above 20 kc, the microphone was located rather close to the jet. The microphone was close enough so that the direct sound predominated the reverberant sound but it was still far enough away to be located outside of the near-field region at these high frequencies. The acoustical measurements with the bare 1/2" diameter nozzle were quite reproducible but when a fine screen was introduced close to the nozzle, a 6 to 8 db discrepancy was observed among the results from repeated experiments. The discrepancy was traced to the rotational

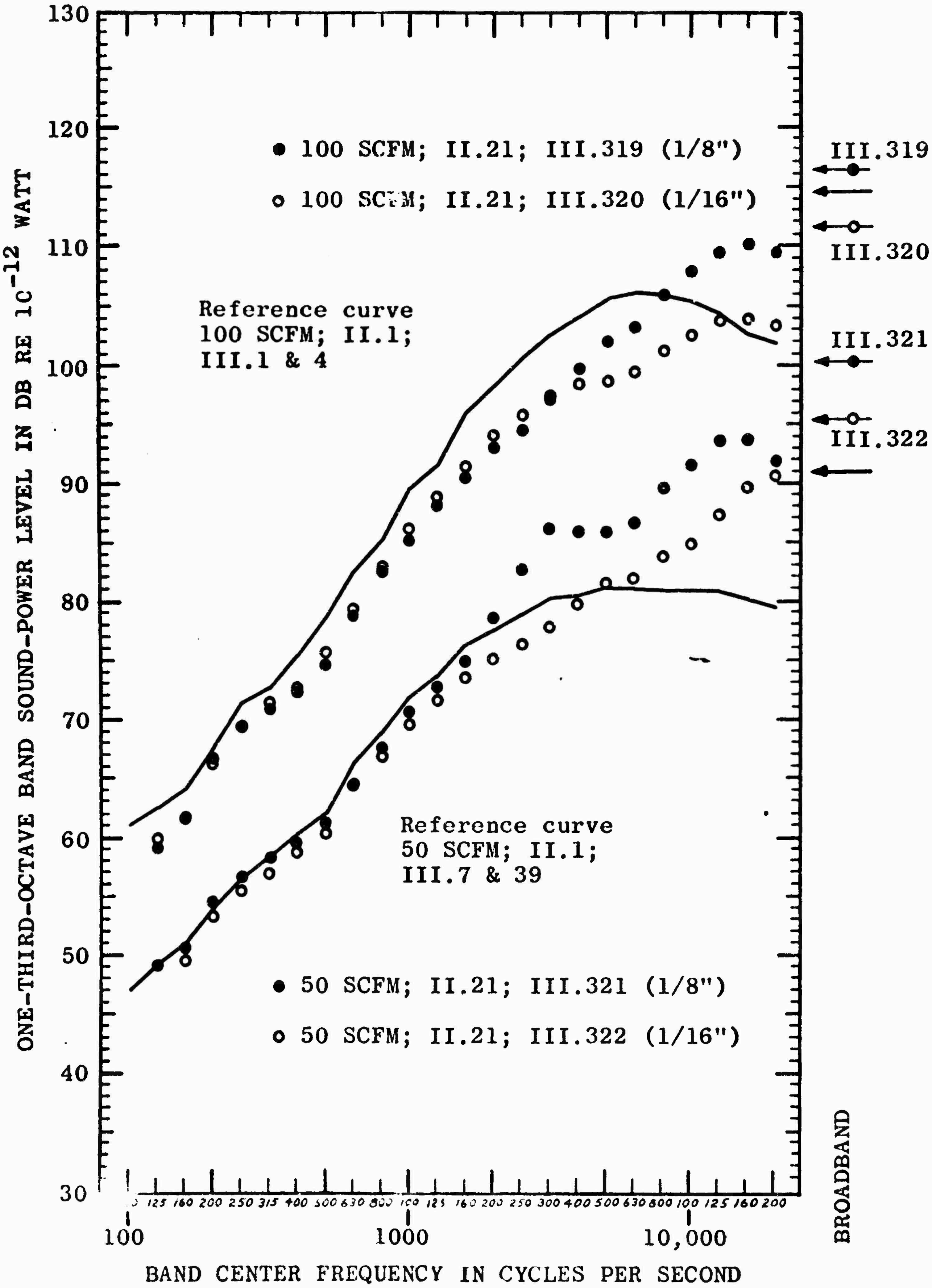

FIGURE 54. SINGLE WIRES ACROSS CENTER OF JET EXHAUST.

orientation of the screen around the jets' axis. That is, the results depended upon whether the warp or the woof lay in the plane containing the microphone and the jet axis. When any particular orientation of the screen was preserved, the noise measurements repeated very closely.

These observations are difficult to reconcile with the small dimensions of the screen's wires and spaces compared to the wavelength of sound or with any obvious asymmetry of the screens or their mounts. Nevertheless, the effect did occur, its magnitude was fully as large as some of the reductions reported in Reference 18 for example, and it could certainly complicate free-field measurements in a most unsuspected manner. With respect to this report, the diffusion in the reverberation room was more than adequate to integrate directionality effects of similar type, if indeed they did occur throughout the frequency range reported, namely 100 cps to 20 kc.

5.4. METAL FELTS

It is fairly evident that screens might constitute only a particular realization of a general silencing principle. If a screen is considered to be a flow resistance or a flow scatterer (and it is not at all certain that either concept is a correct explanation of the observed behavior of screens), then the distribution of this property along the axial direction might improve the acoustical performance. Probably the most obvious experiment is to try several screens arranged in series. This was done and some of the results, which were not especially encouraging, are presented in a later section of this report. However upon further consideration, a series of screens appears to represent an even more specialized three-dimensional distribution rather than a more general three dimensional array of properties. A single screen alone represents a highly-organized two dimensional array and a second screen poses problems of how to locate its openings with respect to those of the first screen. Indeed, the regularity of a single screen may be acoustically inappropriate.

When one seeks to translate the concept of a three-dimensionally randomly distributed flow resistance or flow scatterer into experimental hardware, material problems arise. There doesn't seem to be much available to chose from. There are materials available in the forms of sintered metal powders or metal shot, frittered glass, etc. but these appeared not sufficiently porous and possessed much too small a pore size to be attractive experimentally. A material more akin to steel wool was needed. Steel wool itself has been employed in some types of commercial air-vent mufflers but since it has little or no structural strength, it didn't appear experimentally attractive either. Moreover, it did not seem prudent to try to develop a material or to have it developed within the scope of this contract especially since the desired characteristics were vague.

At this point, an advertisement for a commercial product called Feltmetal came to our attention.[12] The coarsest version of this product offered in sample quantities represented a possible material for experimental purposes and several samples were purchased. This material resembles a textile felt in that it consists of metal fibers laid down randomly and interlocked. Then the material is sintered so that the fibers are joined together to some extent where they touch one another. Thus the finished product possesses appreciable mechanical strength although it looks superficially like a layer of steel wool. Various amounts of metal fibers can be compacted into sheets of different thickness thereby providing for a controlled variation of average porosity. The particular material tested was fabricated from type "B" fiber which is shredded type 430 stainless steel with a mean fiber diameter of approximately 0.004 inch. Additionally this Feltmetal is specified in terms of thickness after sintering and its percentage density, that is, the weight of the finished metal felt compared to a solid piece of stainless steel having the same gross dimensions. Table 4 indicates the properties of the samples which were obtained as well as the nominal value of flow resistance (from manufacturer's data) for each sample.

TABLE 4

FLOW RESISTANCE OF METAL FELTS

($dyne/cm^3/sec$)

Density,	Thickness, inch				
%	1/16	1/8	1/4	1/2	1
5	---	1.0	2.0	4.0	8.0
10	---	2.0	4.0	---	---
20	3.3	6.6	---	---	---

Moreover, on the basis of average pore size, again taken from manufacturer's data, these materials would correlate with a 20 to 25 mesh screen.

The samples were selected originally to allow comparison of equivalent magnitudes of flow resistance for samples of different thickness and density. The sound power measurements promptly indicated that the best silencing results were to be obtained with the 5% materials and so not all of the higher density samples were investigated. Possibly even coarser and more porous

[12] Huyck Metals Dept., Huyck Corp., P. O. Box 30, Milford, Connecticut.

metal felts should be investigated but the appropriate material samples were not immediately available. It should also be remembered that the concept of a three-dimensional distribution of flow resistance has not been investigated generally but only in terms of one available commercial product.

With metal felts as with screens, one encounters an enormous range of parameters which ought to be investigated. Most of the present research with metal felts had to be limited to experimenting with various thicknesses of 5% dense material located at several distances from the nozzle. Figures 55 and 56 illustrate the acoustical consequences of placing a 1/4-inch thick layer of 5% dense material at various distances from the 1/2-inch diameter nozzle. The felt samples were enclosed in a metal ring which served both to space the felt away from the nozzle by the desired distance and to prevent side flow from behind the felt. The felt samples were simply supported at their outer edges and so they had to withstand the air blast from the nozzle.

It is evident from Figures 55 and 56 that the largest reductions in sound power occur with the felt directly against the nozzle; an arrangement which naturally increases the back pressure somewhat. When the metal felt was placed at $x/d \geq 1/4$, the back pressure remained unaffected. The spectra for only three distances are plotted in Figures 55 and 56 although experiments were conducted at other distances also. When the metal felt was moved out of contact with the nozzle face, a significant decrease in effectiveness occurred at all frequencies. For small values of x/d, say from 1/4 to 1.0, the acoustical behavior changes only slightly and closely resembles the spectrum for $x/d = 1/2$ which is plotted in Figures 55 and 56. Still larger values of x/d occasion increased noise accompanied by evidence of tonal generation as well. The semiclosed cavity formed between the nozzle face and the metal felt probably accounts for the tonal characteristics. The same general description of spectral changes with distance applies to both the high and the medium velocity cases although the magnitudes and details vary.

For a thinner layer of metal felt, 1/8-inch thick 5% dense, the variation in acoustical performance with distance was not as pronounced as shown in Figures 55 and 56. The spectra, corresponding to different values of x/d, overlapped at low frequencies while at high frequencies, larger distances resulted in more noise. Thicker layers of metal felt evidenced changes in spectra as a function of distance resembling those shown for the 1/4-inch thick material. The first small separation from contact produced a large change followed by a more gradual and consistent increase of noise power with separation from the nozzle; especially at the higher frequencies.

Often the band levels are not plotted for the lowest frequencies particularly in the cases of the most effective configurations. This stems from the data being background-noise limited for these cases and hence, the experimental values which can be found in Appendix III are not fully representative of the configuration being tested.

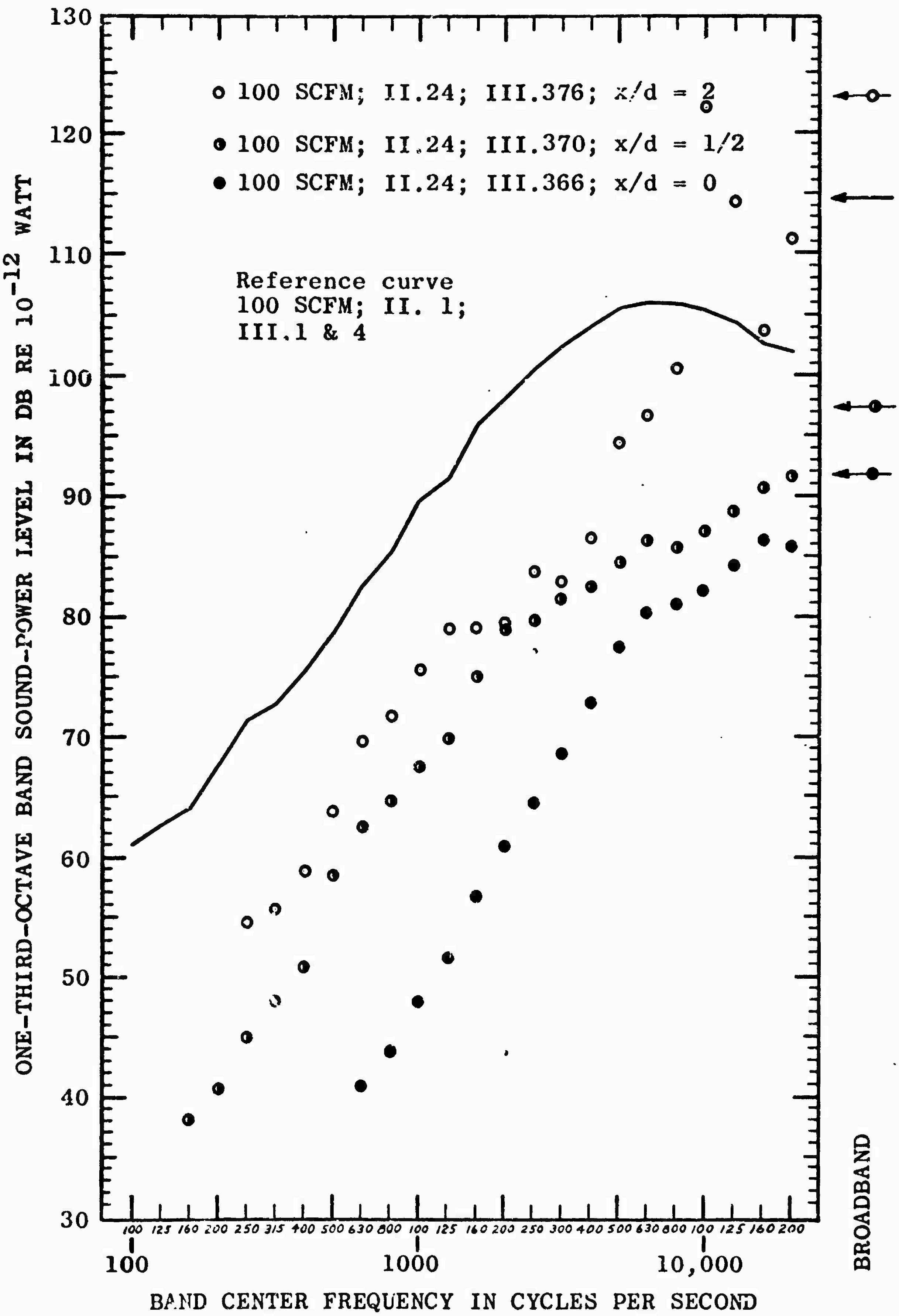

FIGURE 55. EFFECT OF DISTANCE FOR METAL FELT; HIGH VELOCITY.

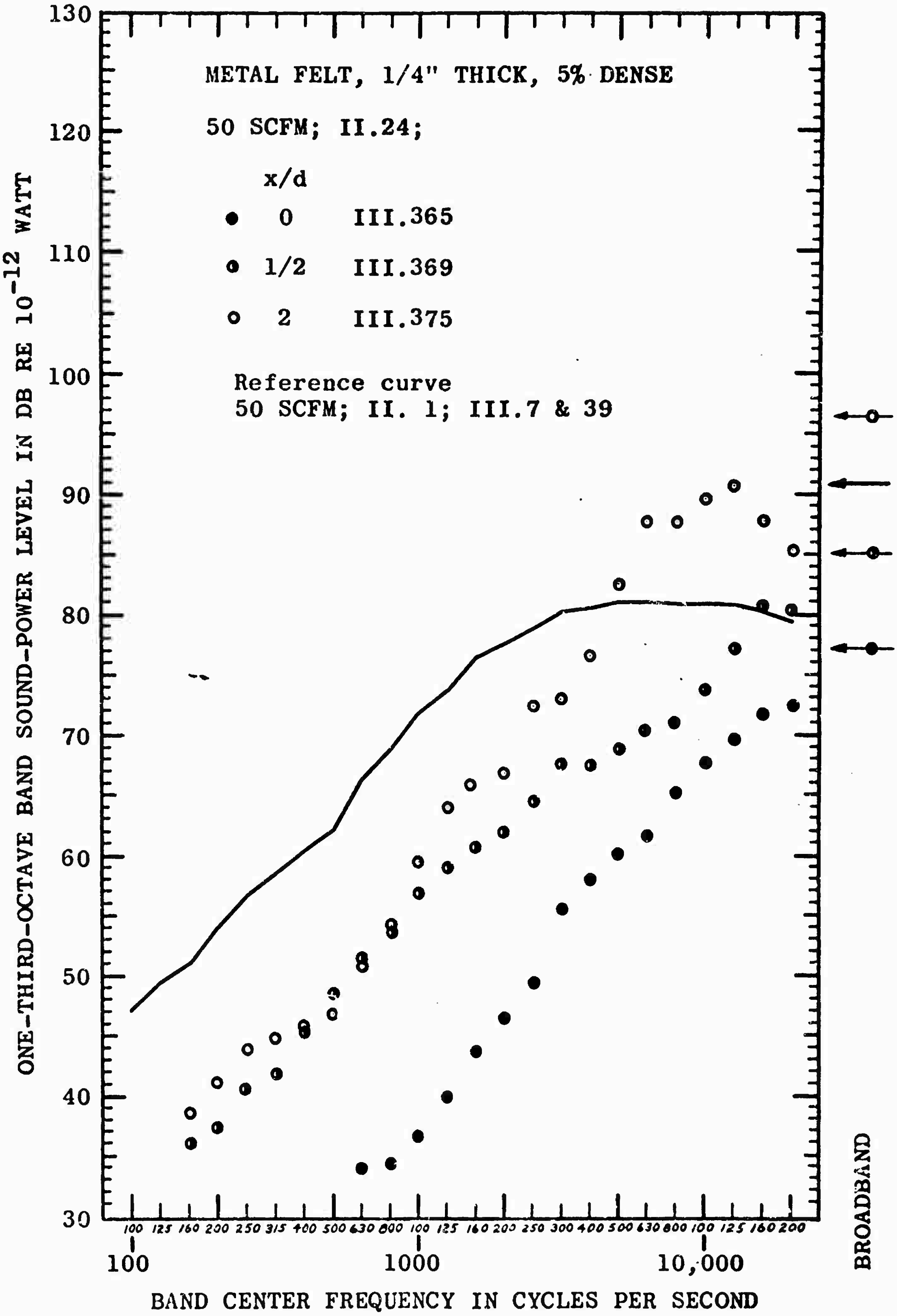

FIGURE 56. EFFECT OF DISTANCE FOR METAL FELT; MEDIUM VELOCITY.

Figures 57 and 58 illustrate the spectral behavior for different thickness of 5% dense metal felt placed at $x/d = 1/4$, that is, 1/8-inch away from the face of the 1/2-inch diameter nozzle. The 1/8-inch thick layer of metal felt exhibited the most noise reduction. Thicker layers of felt produced significant amounts of reduction too but not as much as the 1/8-inch thick layer. An exception occurs at the lowest frequencies in Figure 57 where a cross over appears at about 630 cps. Generally the largest reductions in sound power occur for the highest initial velocity just as it did for screens.

A similar behavior to that illustrated in Figures 57 and 58 was found for larger values of x/d, taking into account the general trends with distance as displayed in Figures 55 and 56. In every instance, the 1/8-inch thick 5% dense felt displayed superior performance. Thicker layers produced results which tended to cluster at an intermediate level of performance and which were comparatively insensitive to thickness also.

Higher density felts yielded markedly less quieting. Figures 59 and 60 compare the spectra for 5% and 20% dense felts having nominally identical thickness. Generally similar behavior was found for larger values of x/d spacing. A slight cross over is observed again in Figure 59 below 500 cps.

It might be inferred from Table 4 that flow resistance could be an important parameter. Since the flow resistances for 1/8-inch thick 5% and 20% dense metal felts are nominally 1.0 and 6.6 dynes/cm^3/sec respectively then perhaps the comparisons drawn in Figures 59 and 60 are not very appropriate. Only a few tests were conducted with the 10% dense felts and so the closest comparisons on the basis of flow resistance as given in Table 4 are not available. However, Figure 61 compares the 1/8-inch thick 20% dense felt (flow resistance of about 6.6 dynes/cm^3/sec) with the 1/2-inch and 1-inch thick 5% dense felts (flow resistance of about 4.0 and 8.0 dynes/cm^3/sec, respectively) which bracket the desired value. Clearly the correspondence is improved over that shown in Figures 59 and 60 but it is equally clear that flow resistance alone is not sufficient. Physical intuition also would suggest that flow resistance probably is only very indirectly related to silencing.

While on the subject of flow resistance, an interesting comparison can be drawn between a screen and a metal felt. Reference 24 relates flow resistance values for several screens and the 100 mesh screen with 0.0045-inch diameter wire seems identical to our 100 mesh screen described in Section 5.3. Reference 24 gives a flow resistance of 9.0 MKS rayls which converts to 0.90 dyne/cm^3/sec. Thus in terms of flow resistance this 100 mesh screen and the 1/8-inch thick 5% dense metal felt are very nearly alike, 0.90 vs 1.0 dyne/cm^3/sec. Moreover, the fibers in the metal felt are approximately 0.004-inch across which compares favorably with the 0.0045-inch diameter wire in the screen. Figure 62 compares the sound power spectra for these two materials both located at $x/d = 1/4$. The metal felt obviously generates less noise than the screen all across the spectrum; less by roughly an order of magnitude.

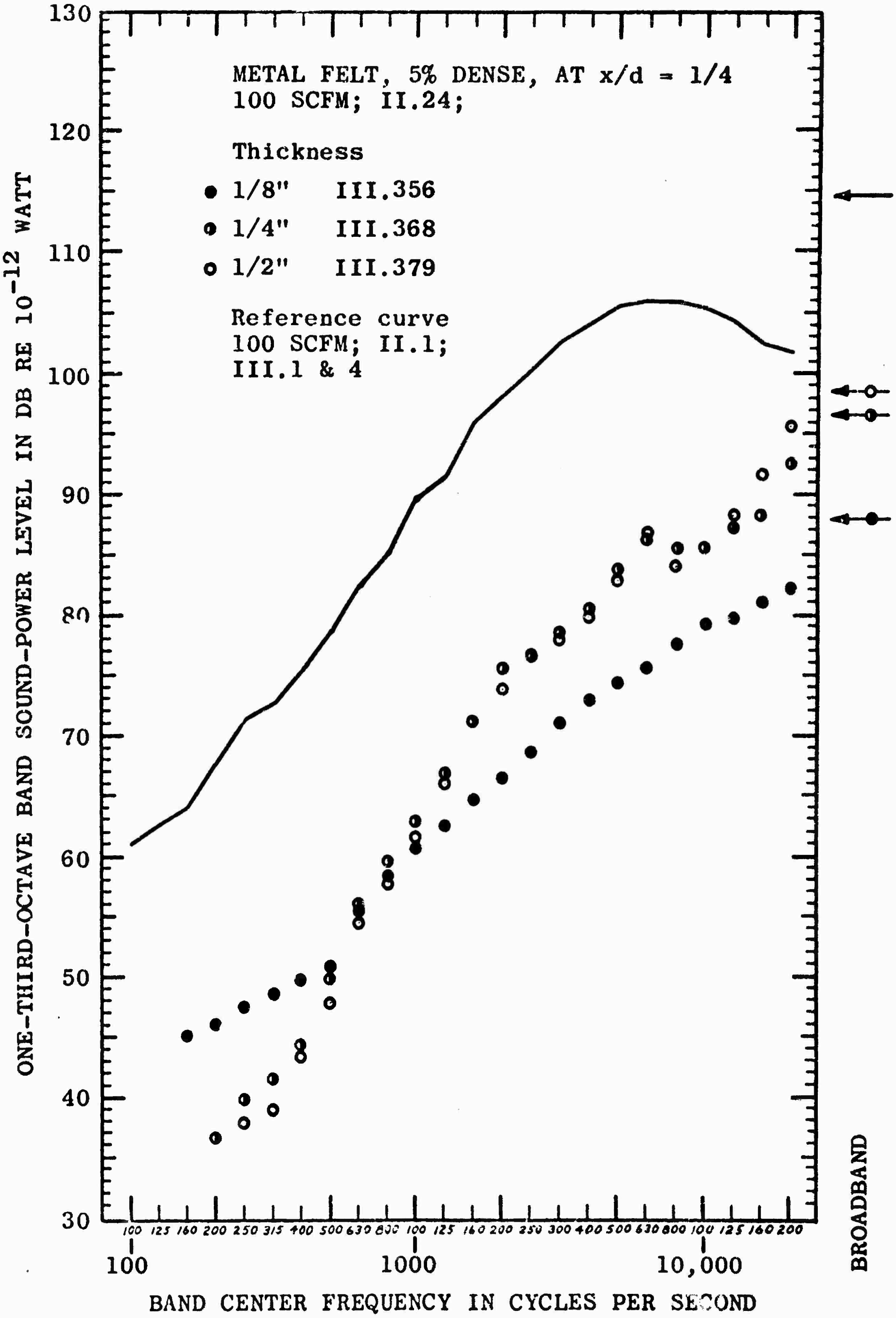

FIGURE 57. EFFECT OF METAL FELT THICKNESS; HIGH VELOCITY.

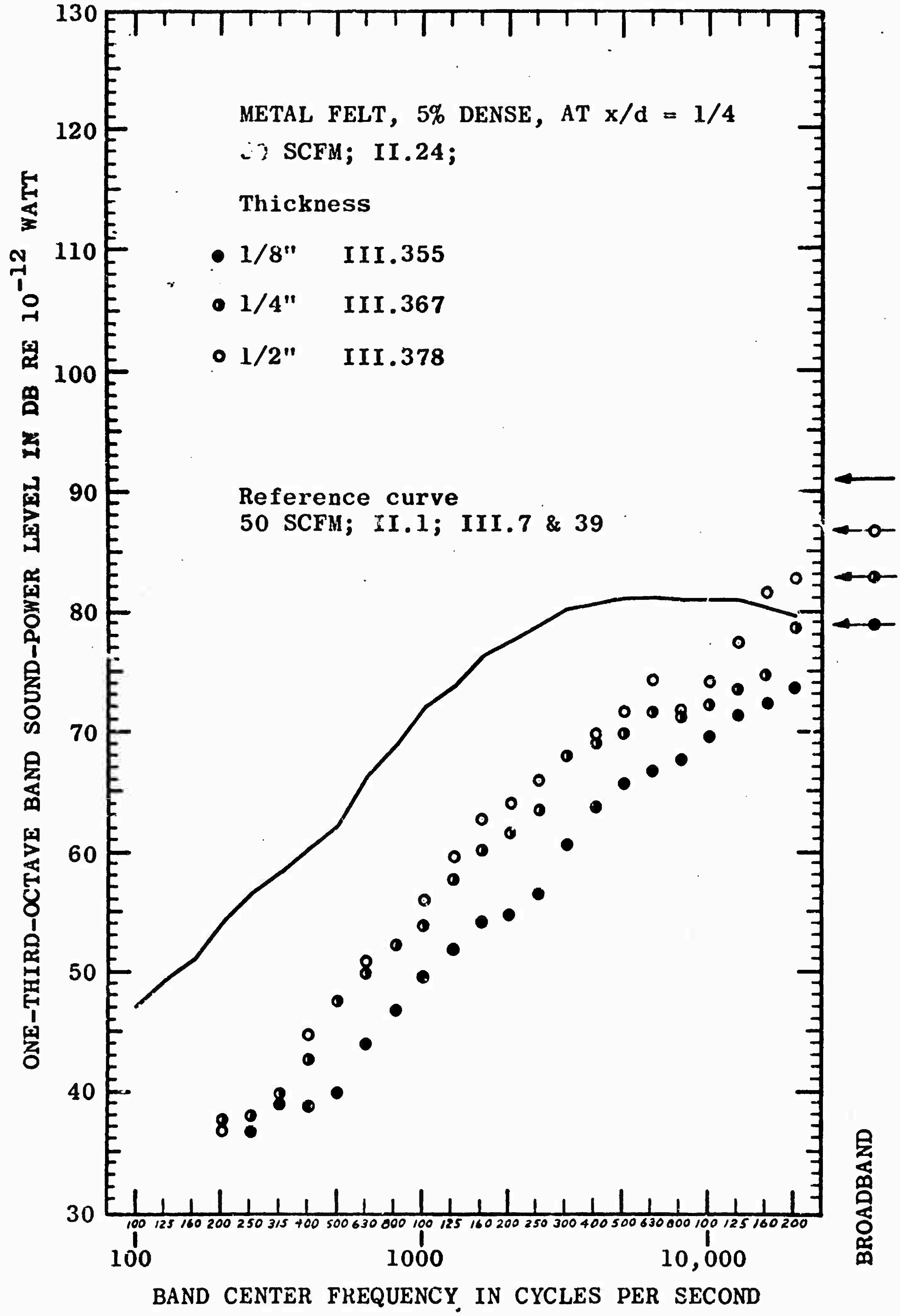

FIGURE 58. EFFECT OF METAL FELT THICKNESS; MEDIUM VELOCITY.

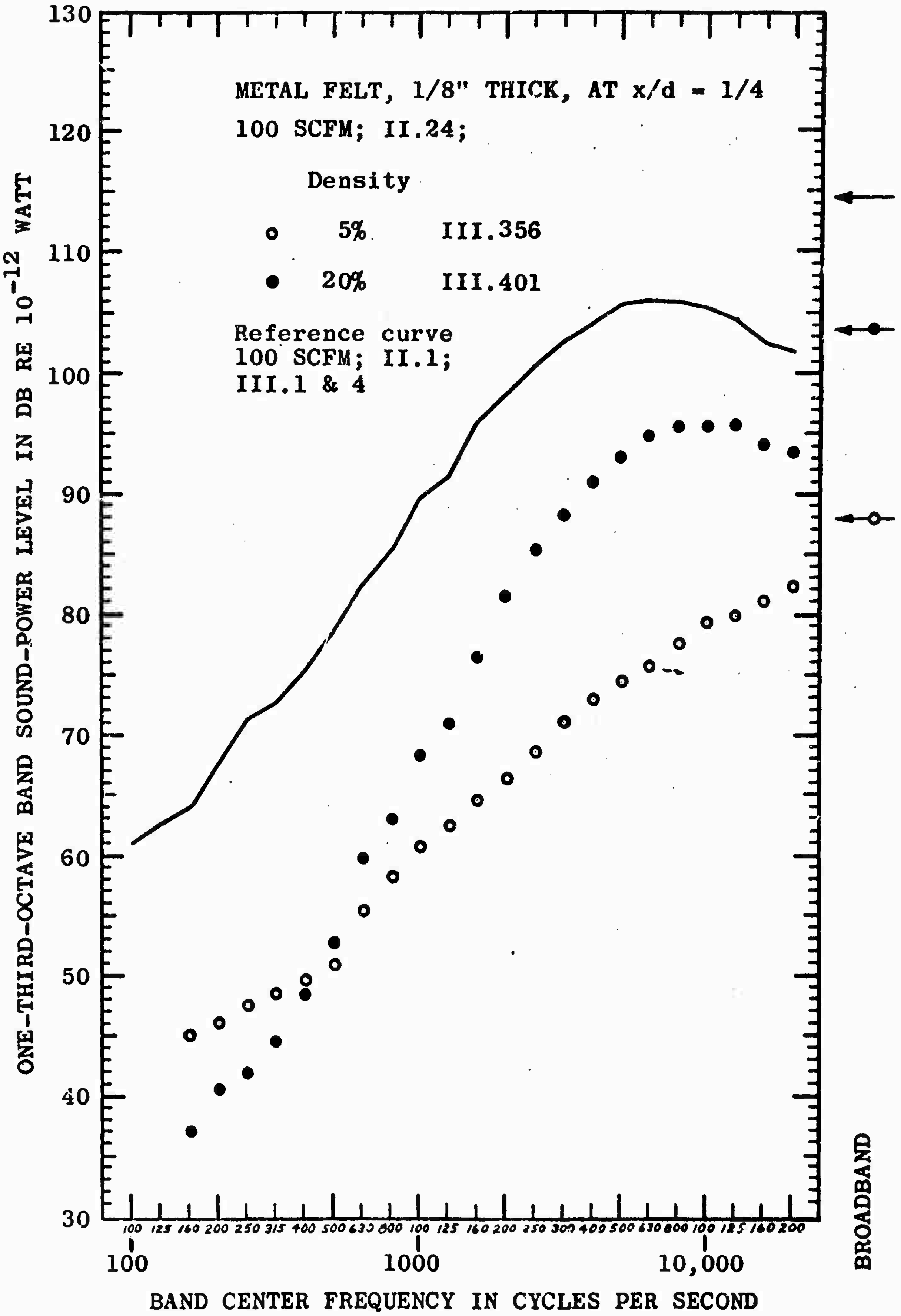

FIGURE 59. EFFECT OF DENSITY OF METAL FELTS; HIGH VELOCITY.

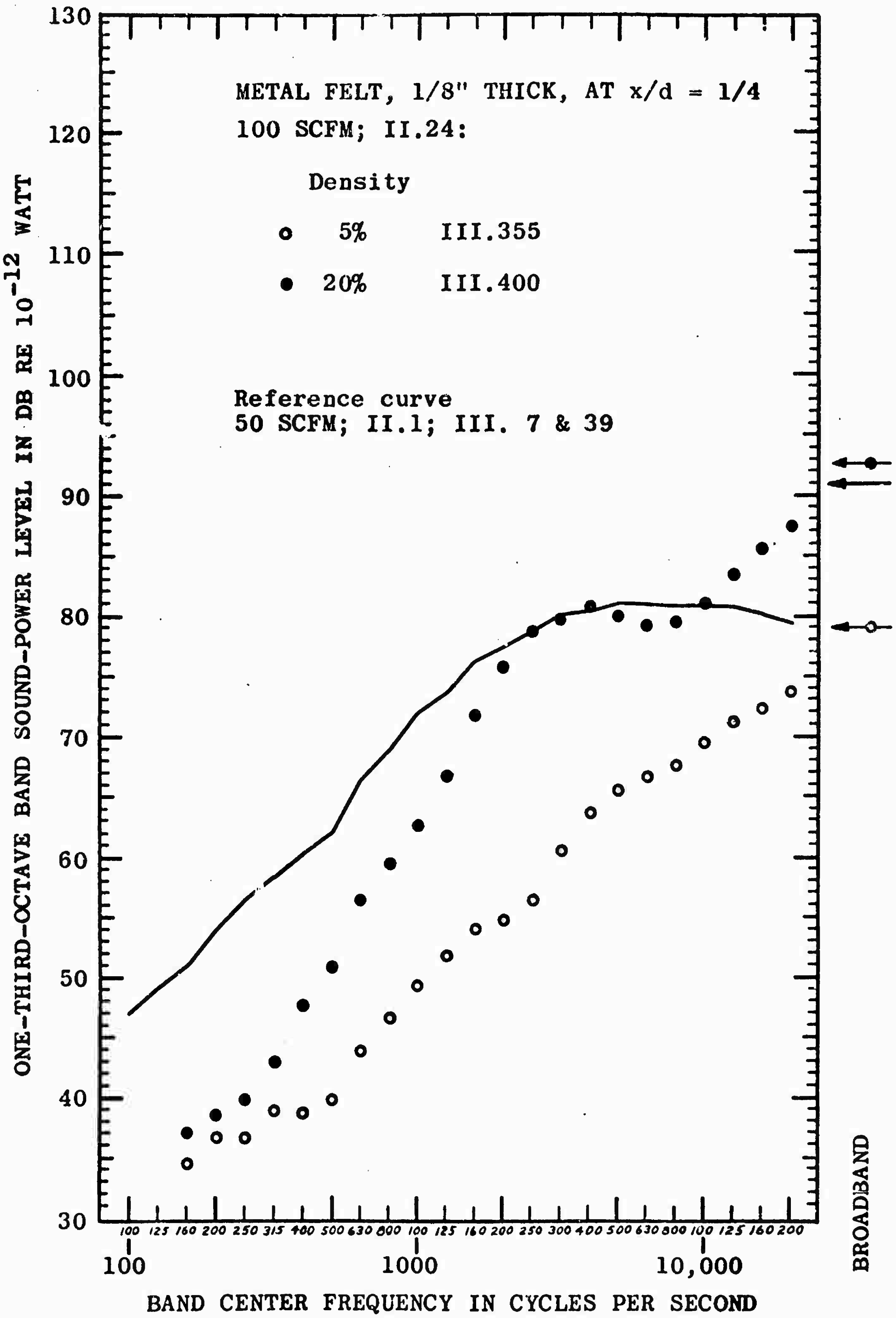

FIGURE 60. EFFECT OF DENSITY OF METAL FELTS; MEDIUM VELOCITY.

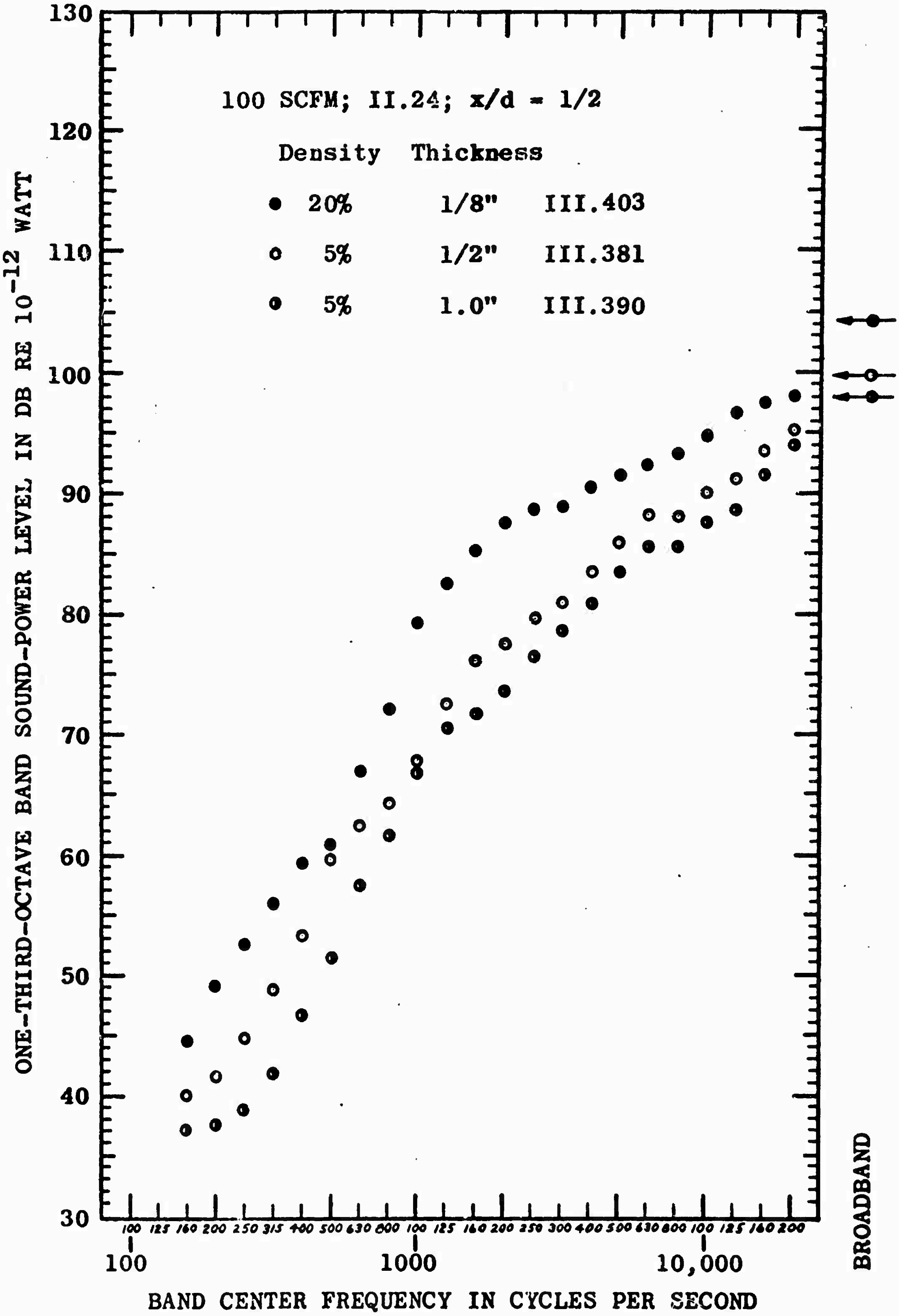

FIGURE 61. COMPARISON OF FELTS FOR SIMILAR FLOW RESISTANCE.

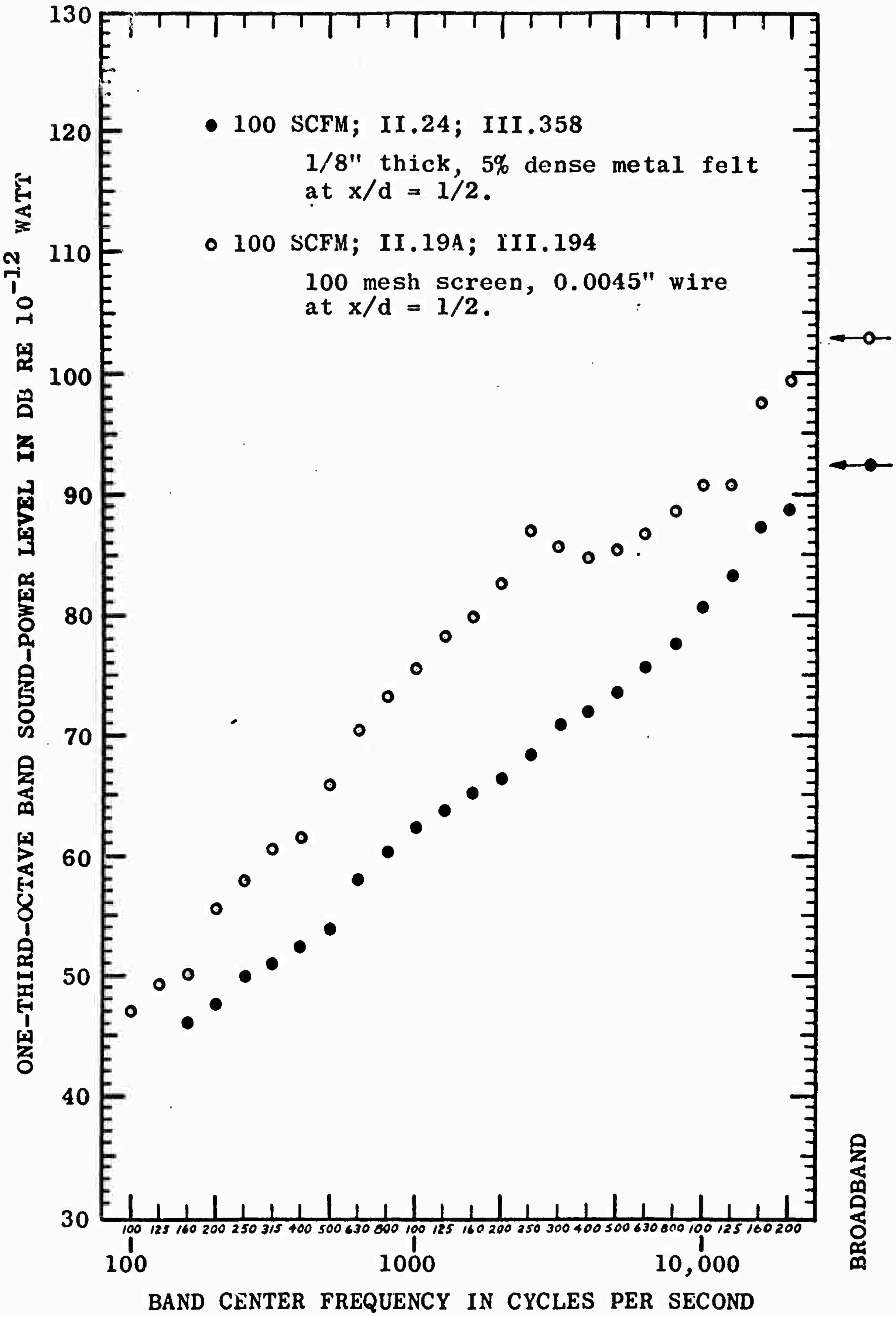

FIGURE 62. COMPARISON OF A METAL FELT AND A RELATED SCREEN.

A metal felt is visibly different from a screen in several ways and it is not clear how these relate to the acoustical performance. A screen is essentially two-dimensional whereas the felt has appreciable thickness. The experiments reported here have indicated that thick layers (of constant density) of felt are comparatively noisy but an optimum value of small thickness has not been demonstrated. Screens possess a regular structure implied by the term mesh whereas felts have a random structuring which needs statistical description. The regularity of a screen might contribute an acoustical coherence which would be absent in the case of a felt. The metal fibers of the felts used in these experiments appear to have been formed by shearing and thus have irregular cross sections while the screen was fabricated from round wire.

One can also formulate a plausible explanation by considering the felt to approximate a random scatterer which disorganizes the jet efflux into the random motion of a slowly diffusing gas. Whatever the explanation in microscopic detail, there is an element of effectiveness associated with a thin layer of low-density felt which ordinary screens do not possess.

SECTION 6

EFFECTS OF COMPOSITE ARRANGEMENTS

This final major section of the report is concerned with the acoustical effects which accrue when combinations of objects are used to influence the noise from jets. Preceding sections have described silencing tests utilizing a variety of objects tested one at a time. It would have been ideal if those tests could have been exhaustive but obviously they could not in a program of moderate size. In this section, several systematic experiments with combinations of objects have been undertaken to ascertain if and how such objects may be combined to yield more silencing. Again exhaustive research has not been possible but several combinations have been investigated to the point where their possibilities have been moderately-well delineated. Several miscellaneous experiments are also described.

6.1. MULTIPLE SCREENS

Since screens constituted the first class of objects exhibiting much promise of useful noise reduction, questions about the acoustical effectiveness of screens arranged in series occur naturally. In general, these multiple screen experiments yielded negative results or results not significantly better than for single screens. A hint of this general behavior has appeared already in the fact that the acoustical effectiveness of any particular single screen decreased with lower initial velocities at the nozzle, see Section 5.3.

However, if the first screen is coarse, open, and located rather close to the nozzle and if a second finer screen is located slightly farther downstream, then a larger noise reduction can be obtained than for either screen alone. Moreover, the net reduction can be somewhat larger than for the best single screen configuration. Figures 63 and 64 illustrate the silencing produced by a 30 mesh screen and a 50 mesh screen in series. To aid comparison, the spectra representing the 30 mesh screen alone have been superimposed. The spectra for the 50 mesh screen alone have been omitted but these are similar to the spectra for the 30 mesh screen alone.

At medium velocity and at high frequencies (see Figure 64), neither the single screen nor the double screens contributed silencing. If a simple double-screen muffler were contemplated, additional research would be required to find out if this high-frequency behavior can be cured.

The addition of more layers of screens gave no useful results. While it cannot be claimed that all possible compounding of screens has been thor-

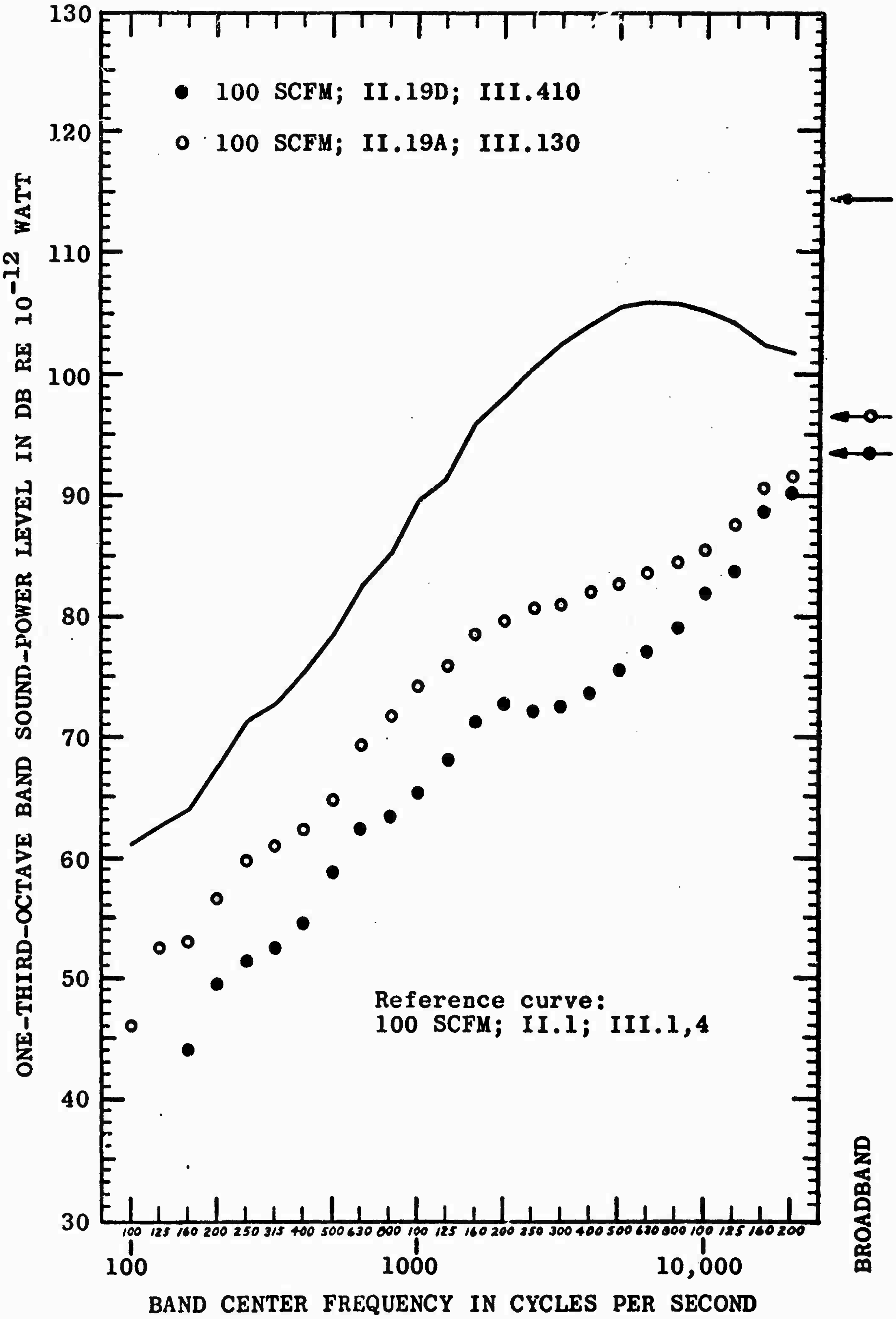

FIGURE 63. EFFECT OF TWO SCREENS IN SERIES; HIGH VELOCITY.

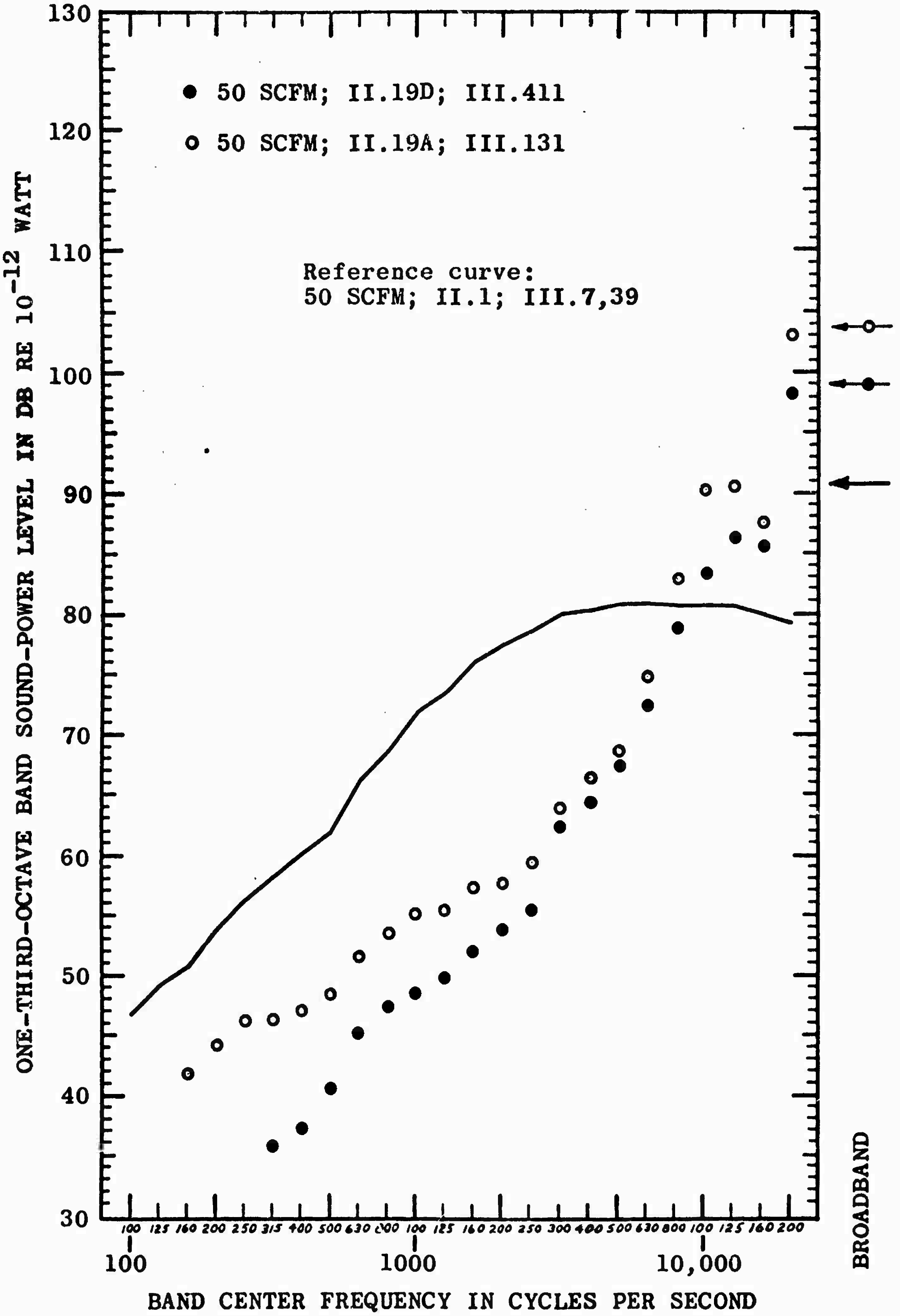

FIGURE 64. EFFECT OF TWO SCREENS IN SERIES; MEDIUM VELOCITY.

oughly investigated, it looks doubtful if much more can be gained by continuing in this direction. Possibly by critical adjustment of screen parameters, perhaps starting with a higher percentage of open area, some useful gains could be realized. As stated earlier, however, a single screen (or now perhaps a double screen) hold acoustical promise of being developed into a very simple muffler of moderate effectiveness. Thus the findings of this research generally confirm and supplement the NACA work reported in Reference 19 although some disagreements among the details remain. The material selection and/or engineering problem with respect to screens has not been a major concern of this research and so a final choice of materials with respect to temperature and durability may constitute a formidable problem.

At one stage during the experiments with multiple screens, it was postulated that downstream of each screen it might be necessary to reform a single jet at lower velocity before introducing the next screen. Experiments in this direction gave no improvement over the double screens already described. Indeed, multiple chamber acoustical effects occurred as might be expected and so this course of research was not pursued further.

6.2. METAL FELTS AND SCREENS

The investigation of metal felts as silencing devices could have proceeded in several directions. Previous experiments had demonstrated that a thin layer of low-density felt was more effective than thick layers. Possibly a felt of low mean density but with density increasing in the downstream direction might have useful acoustical properties but materials with such characteristics were not available without undertaking a concomitant materials development.

The sintering process endowed the metal felts with appreciable rigidity but the air-flow forces were still too large for the thinner felts under some test conditions. The mechanical failures of some of the felts raised the question of whether a mechanically-strong screen could be used to reinforce a comparatively weak felt. The first experiments in this direction demonstrated significant acoustical interactions and so a rather extensive set of related experiments was undertaken. (See III.412-525)

Figures 65 and 66 are generally indicative of the results for the experiments in which a screen supported the metal felt. As a point of departure, the 1/8-inch thick 5% dense felt was located at $x/d = 1/4$ which had been found to be an optimum location. This case is represented by the solid dots in Figures 65 and 66. When the layer of metal felt was supported by a 10 mesh screen, the sound power was increased slightly all across the spectrum as shown by the open circles in Figures 65 and 66. When the felt was supported by a 20 mesh screen, the spectrum was scarcely different than for the felt without support; this condition has not been plotted. When finer screens were used for support, the radiated sound power was reduced all across the spectrum.

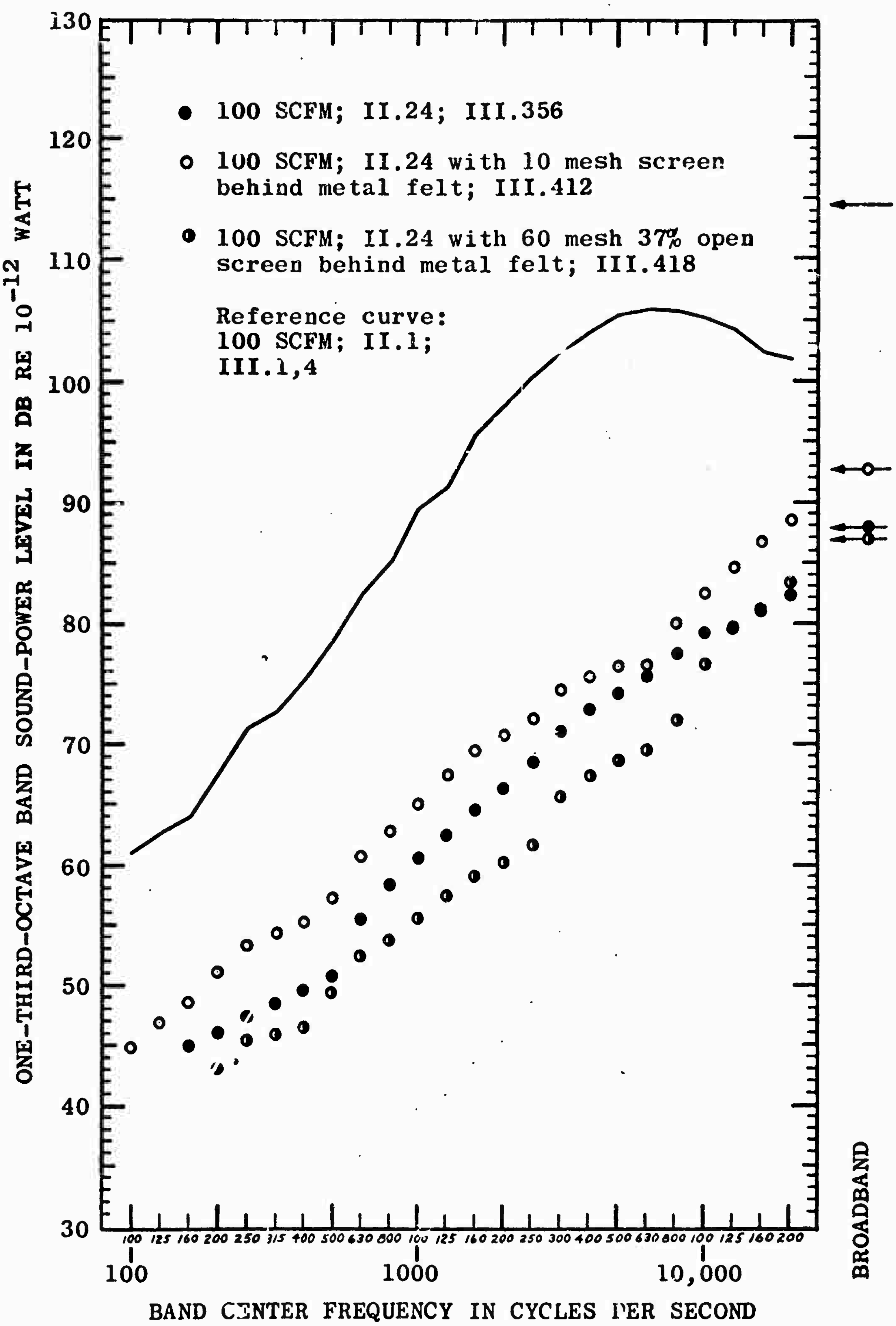

FIGURE 65. SUPPORT OF FELTS BY SCREENS; HIGH VELOCITY.

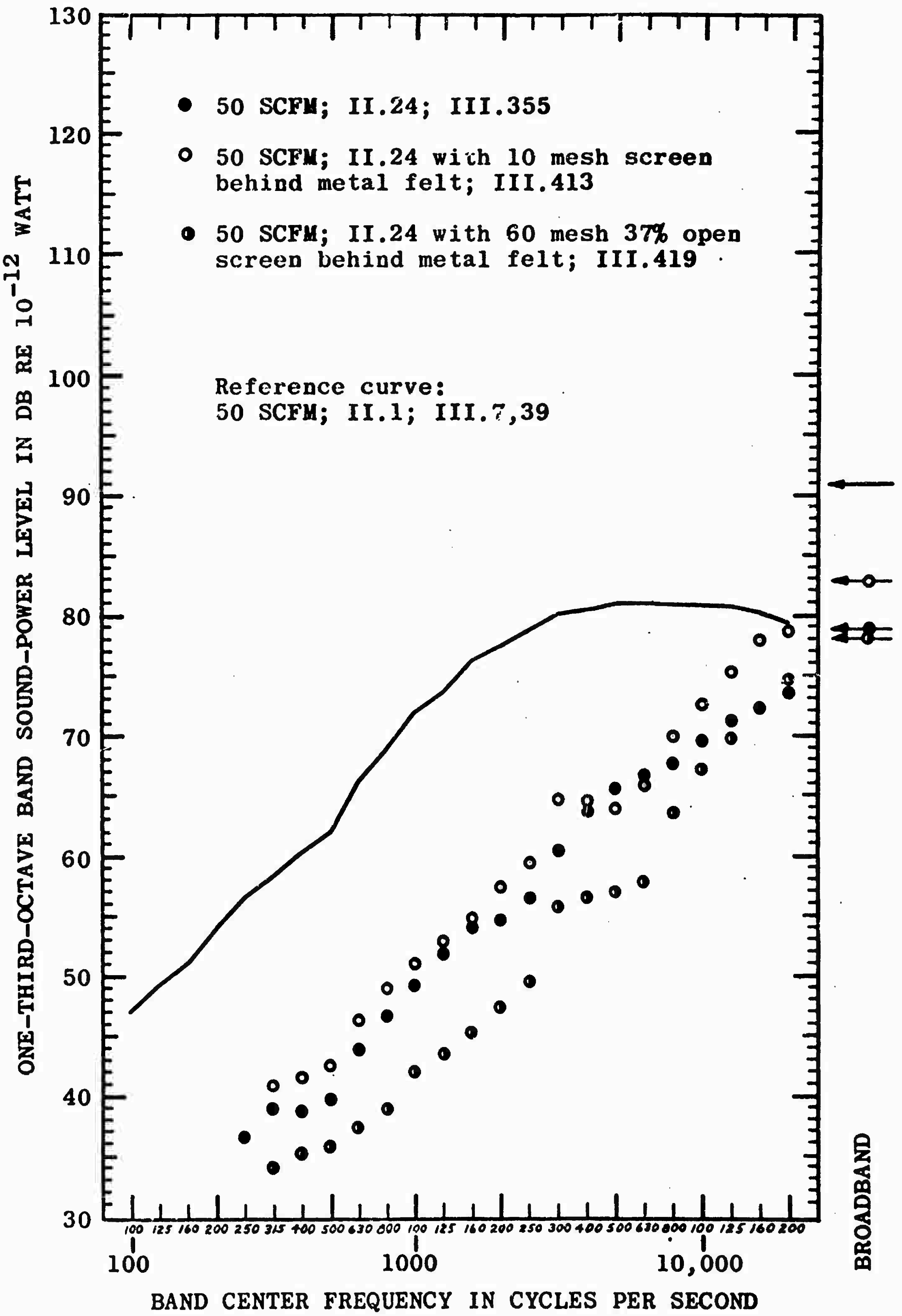

FIGURE 66. SUPPORT OF FELTS BY SCREENS; MEDIUM VELOCITY.

The results for the 60 mesh 37% open screen are indicated by the half-open dots in Figures 65 and 66. The 100 mesh 30% open screen yielded results nearly identical to those for the 60 mesh 37% open screen. In the case of the 400 mesh screen, somewhat more noise was generated over most of the spectrum but not as much as for the metal felt alone. Also, the 60 mesh 16% open screen consistently produced more noise than the 60 mesh 37% open screen when used to support the metal felt although the differences were small.

There appears to be an optimum mesh for a screen used to support the metal felt and in the experiments just discussed, this optimum occurs in the vicinity of 60 to 100 mesh. These results were obtained with the upstream surface of the metal felt located at $x/d = 1/4$. The same general form of results was obtained when the metal felt was located further downstream at $x/d = 1/2$ and $x/d = 1$ respectively. The principal difference in these cases was that the metal felt alone gave a relatively more intense spectrum so that the addition of a 10 mesh support-screen now caused a slight reduction rather than an increase in the noise. On the whole, the 60 and 100 mesh screens were as good as any and the combination of a metal felt supported by a screen was significantly better than the metal felt without a screen.

Similar experiments were conducted with the 0.707 inch (III.466-501) and the 1.00 inch (III.502-519) diameter nozzles. These experiments confirmed that the most silencing occurred for a relatively close spacing, say $x/d = 1/4$ and that for best results, the supporting screen should not be as fine as 400 mesh. Somewhat coarser screens produced more silencing effect, a fact which suggests that the backing screen may be scaled up for application to full-size jet engines.

The best case illustrated in Figure 65 corresponds to a broad-band noise reduction of roughly 28 db. A large reduction is in evidence across the whole spectrum but the amount of silencing occurring at low-frequencies is particularly interesting. The shape of the silenced spectrum, that of a continuously rising curve, perhaps poses more questions than it answers. Certainly this trend cannot continue to indefinitely high frequencies but not very much more can be deduced from the present experiments. The observed spectral changes may be purely frequency changing resulting from the small pore size however other experiments with screens and metal felts have suggested that frequency changing by itself is not a sufficient explanation. Even in Figure 65, the slopes of the "silenced" spectra are more like +6 db per octave instead of the +9 db per octave to be expected for simple jet behavior below peak frequency.

Figures 67 and 68 contrast locating a screen ahead of the metal felt with locating it behind the felt where it must be to furnish structural support; solid and open dots respectively. The solid line represents the sound power spectrum for the same metal felt without a screen. These spectra are for the metal felt located at $x/d = 1/2$ and so the 20 mesh screen is able to produce some additional quieting. On the average, placing the 20 mesh screen ahead of the felt produced somewhat superior results.

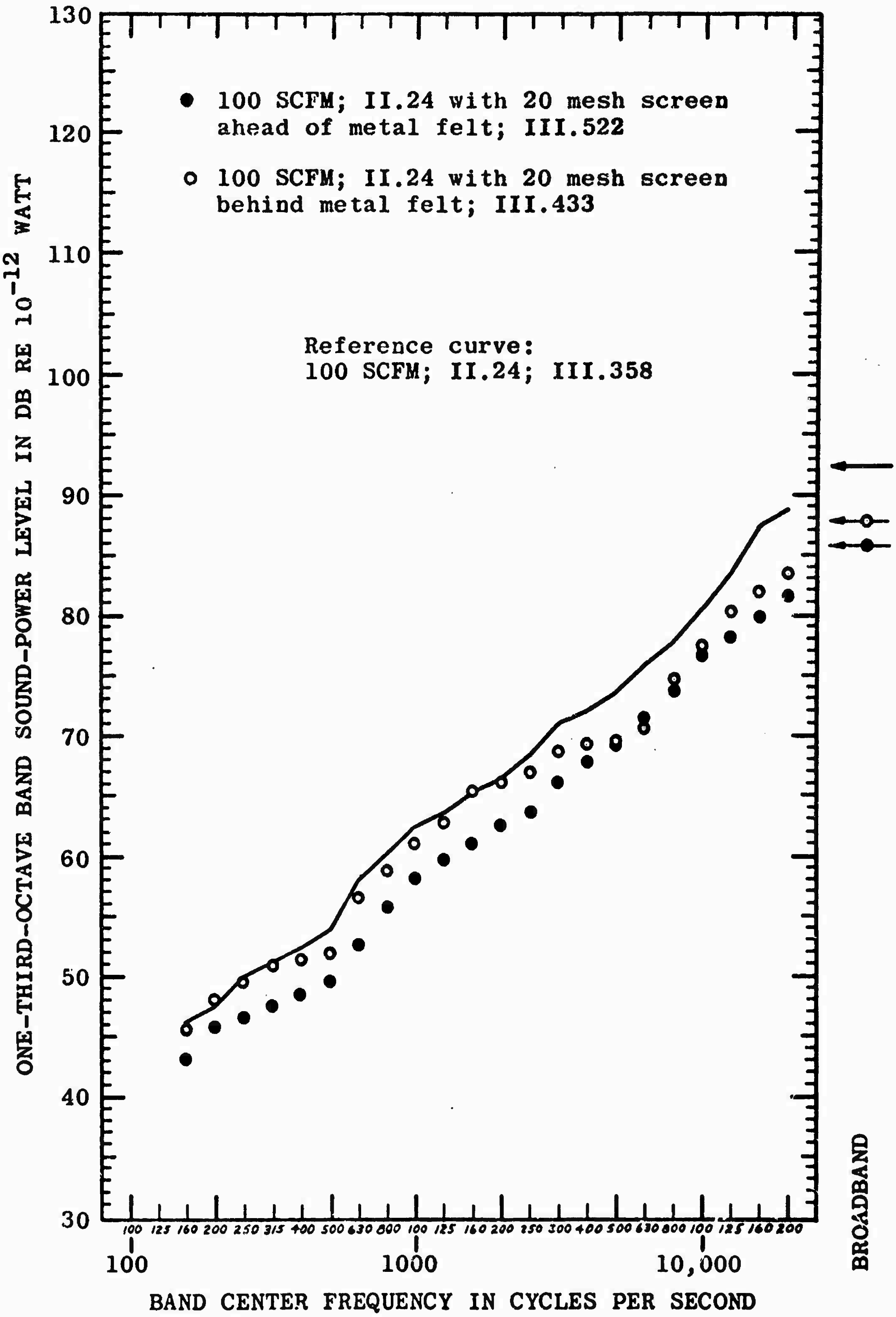

FIGURE 67. EFFECT OF SCREEN LOCATION; HIGH VELOCITY.

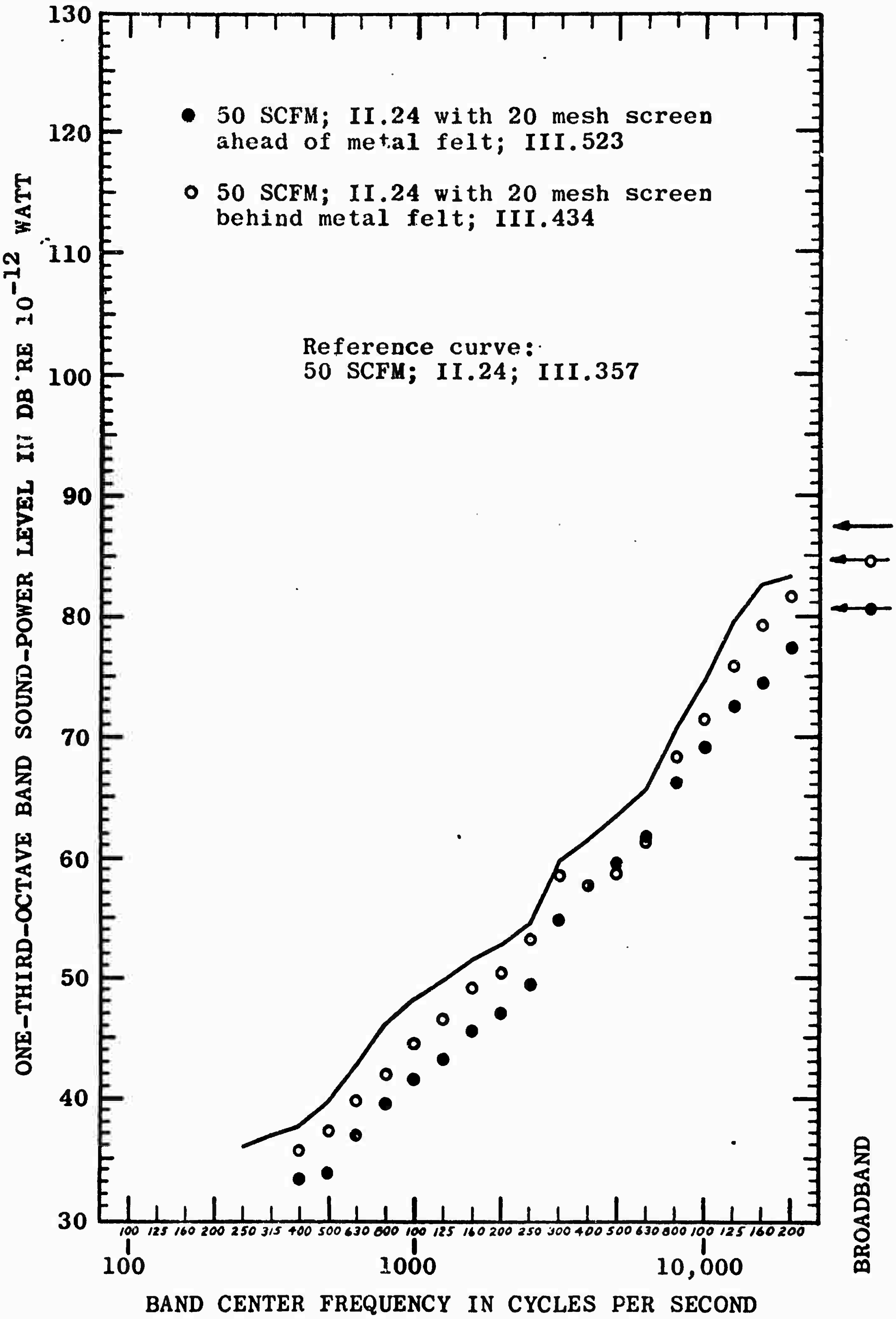

FIGURE 68. EFFECT OF SCREEN LOCATION; MEDIUM VELOCITY.

In Figure 69, an attempt is made to demonstrate the success of the research in terms of the criterion based on the noise generated by simple nozzles as proposed in Section 3.1 and 3.2. The two solid curves represent the experimental spectra for the 0.500 inch and the 1.00 inch diameter smooth-approach nozzles operated at 100 SCFM. The open circles represent the 0.500 inch nozzle silenced by a 1/8-inch thick layer of 5% dense metal felt supported by a 10 mesh screen and located at $x/d = 1/2$. The solid dots represent the same felt-screen configuration followed by a 1/2 inch-long spacer ring and the 1.00 inch diameter smooth-approach nozzle to reform a final jet. The addition of the spacer and nozzle to the felt-screen configuration obviously has increased the low-frequency noise and decreased the high-frequency noise. The latter is probably the result of partially enclosing the noise sources and the former perhaps due to reforming a jet or to creating an acoustical cavity. However, that may be, below 630 cps the black dots fall below the experimental spectrum for the 1.00 inch diameter smooth-approach nozzle alone. This discrepancy may not be surprising since in Figure 5 it had been found that the 1.00 inch diameter nozzle produced more noise than expected for the equivalent simple jet, perhaps due to a flow separation upstream of the nozzle. Thus in the present case, possibly the black dots correspond to a flow condition more closely approaching that of the postulated simple jet so far as the low-frequency noise is concerned. However, a comparison with Figure 4 reopens the question because if the postulated spectrum is matched to the experimental data for the 0.500 inch diameter nozzle then the black-dot spectrum of Figure 69 still dips below the corresponding postulated spectrum. Alternatively, matching the postulated spectrum to black dots at low-frequencies causes a mismatch for the 0.500 inch diameter nozzle which is difficult to reconcile. It is possible that the observed discrepancy is an artifact of the measurements or the test conditions, but it is also possible that the accepted description of simple jet noise is not quite the absolute limiting case which it is usually presumed to be.

Utilizing only metal felts and screens, a silencing configuration was devised which avoided the use of solid, acoustically opaque boundaries. Figures 70 and 71 show the results as solid black dots while the same metal-felt configuration enclosed in rigid tube to produce acoustically-opaque walls gave the open-circle results. When opaque boundaries are avoided, the result is a smoothly rising spectrum and for the high-velocity condition shown in Figure 70, a broadband reduction of about 28 db was obtained. The slope of the spectrum is close to +6 db per octave and, as usual, a question remains about the behavior at still higher frequencies. For the medium velocity test condition displayed in Figure 71, the broadband silencing was not as large and the smoothly rising spectrum acquired a slightly steeper slope of perhaps +7 db per octave.

In both cases, the addition of the solid boundary conditions occasioned significant alterations in the spectra and because the predominent high frequency bands were slightly reduced, so also were the broad band levels. Actually, the solid boundaries are somewhat "shielded" from the noise in the jet by the

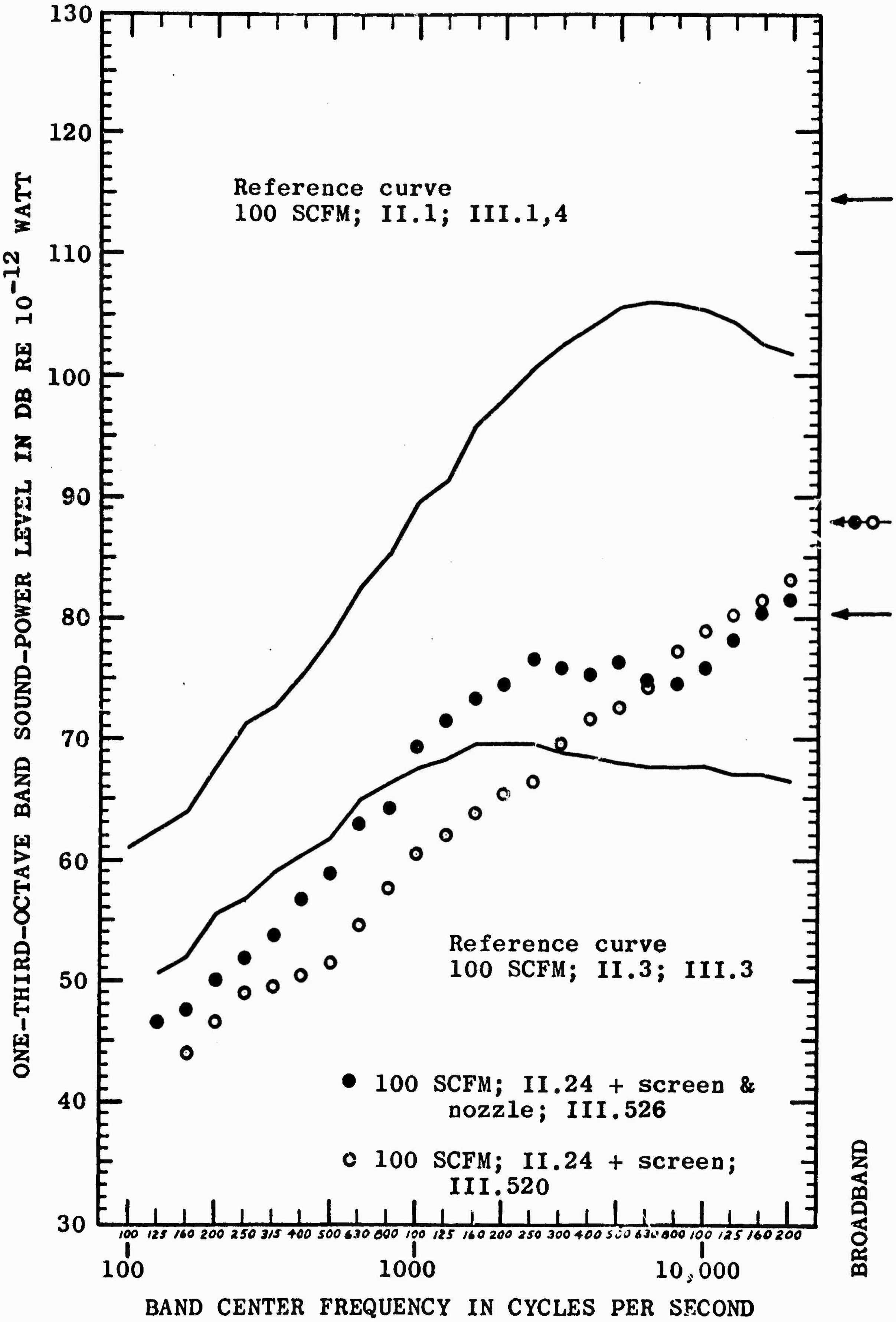

FIGURE 69. FELT-SCREEN COMPARED WITH SIMPLE JET SPECTRA.

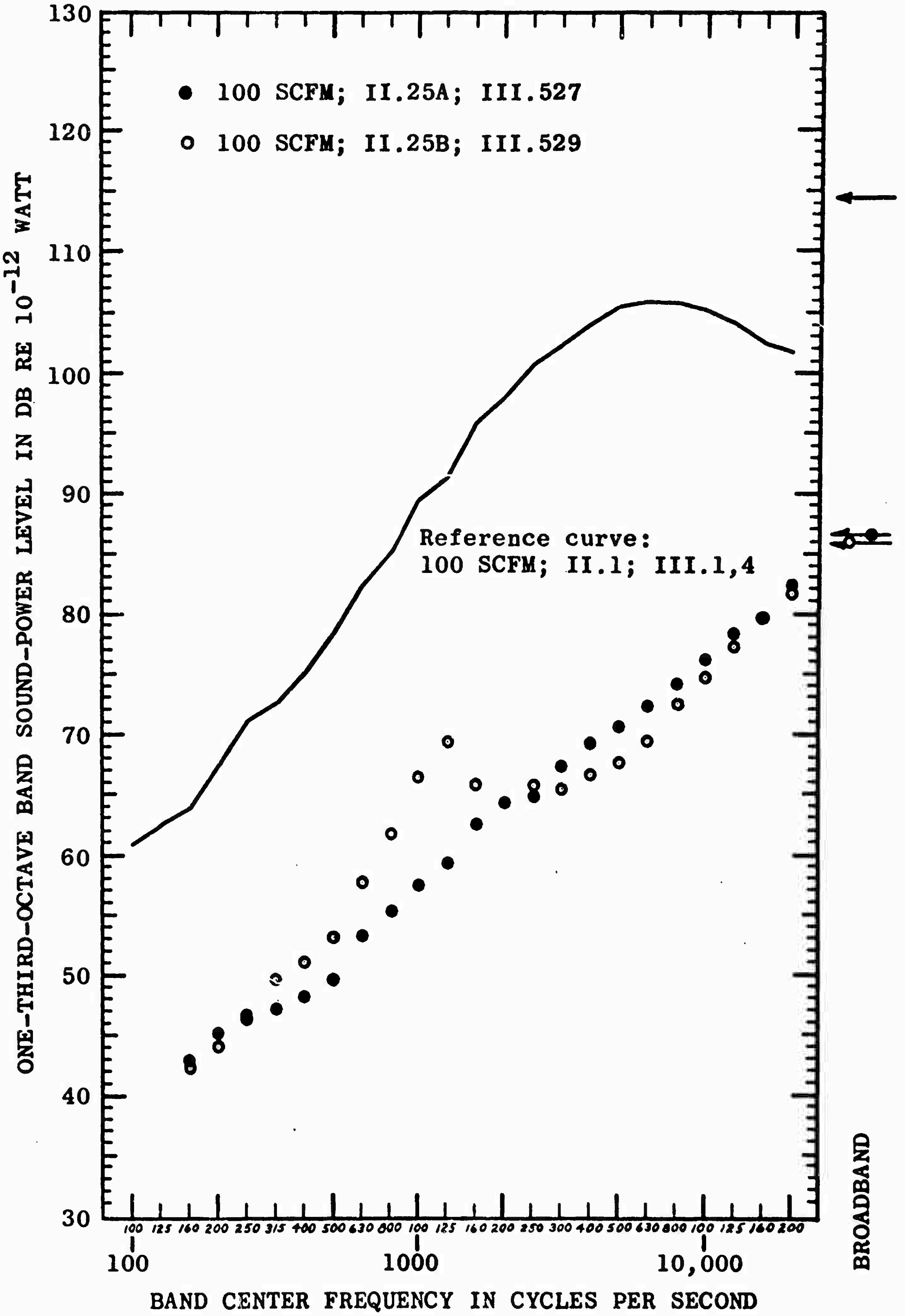

FIGURE 70. METAL FELT-SCREEN CONFIGURATION; HIGH VELOCITY.

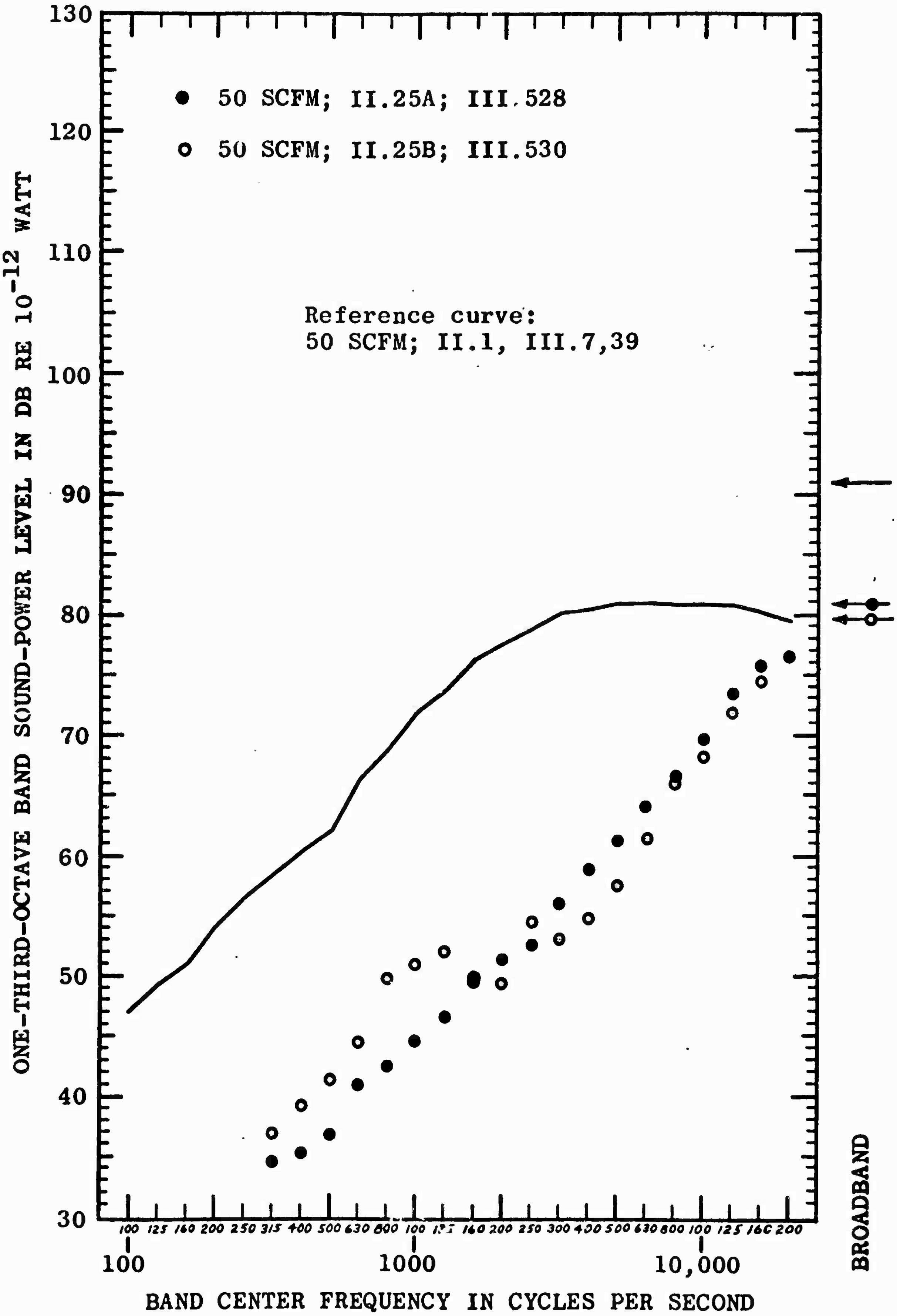

FIGURE 71. METAL FELT-SCREEN CONFIGURATION; MEDIUM VELOCITY.

metal felts and the observed changes in spectrum are probably the net result of several changes in the acoustical processes occurring simultaneously.

Figure 72 repeats the results from Figure 70, now displayed as solid and dashed lines respectively, and shows, by means of solid dots, the effect of adding a terminal 1/8 inch thick layer of 5% dense metal felt supported by a 20 mesh screen. The effect of this terminal layer of metal felt at medium velocities so closely resembled that in Figure 72 that a new graph is unnecessary. The principal result is an appreciable reduction at the lowest frequencies with some alteration of details at higher frequencies. The 20-mesh supporting screen probably was not needed for acoustical reasons but was a convenient means for holding this configuration together.

The comparatively large amount of silencing obtained with the metal felts and screens described above led to a further compounding of the silencing structure with more spacers, layers of metal felts and screens. Each additional stage was chosen somewhat arbitrarily from amongst the hardware at hand. Consequently the resulting configurations have not been optimized and probably contain ineffective components. However Figures 73 and 74 show the results for three such experimental configurations.[13] Appreciable improvement in silencing followed from each increase in configuration complexity. Most importantly, large improvements occurred at high frequencies.

The solid black dots in Figures 73 and 74 represent the largest amount of silencing obtained during the project. For the high-velocity condition shown in Figure 73, the broadband sound power level was reduced by 39 db, that is, to nearly 1/10,000 of its original sound power, while some individual bands were reduced as much as 44 db.

Too little research time remained to investigate how to optimize this configuration, either with respect to components or to dimensions, or to work further on the high-frequency portion of the spectrum. Moreover, the initial concept of avoiding rigid boundaries was dropped and probably the rigid boundaries account for some of the residual bumpiness at mid-spectrum. However, the most effective configuration, just as it existed without any further optimization, would only become 13 feet in diameter by 8 feet long when scaled up linearly by a factor of 40.

6.3. "MUFFLER" BODIES

Several series of experiments have been based upon the use of a right-angled elbow fabricated from metal tubing as shown in II.27. The dimensions and shape were selected to correspond to a crudely-modeled scale version of

[13] These particular configurations were selected by Mr. Philip G. Kessel following his measurements on the configurations of Figures 70 and 71.

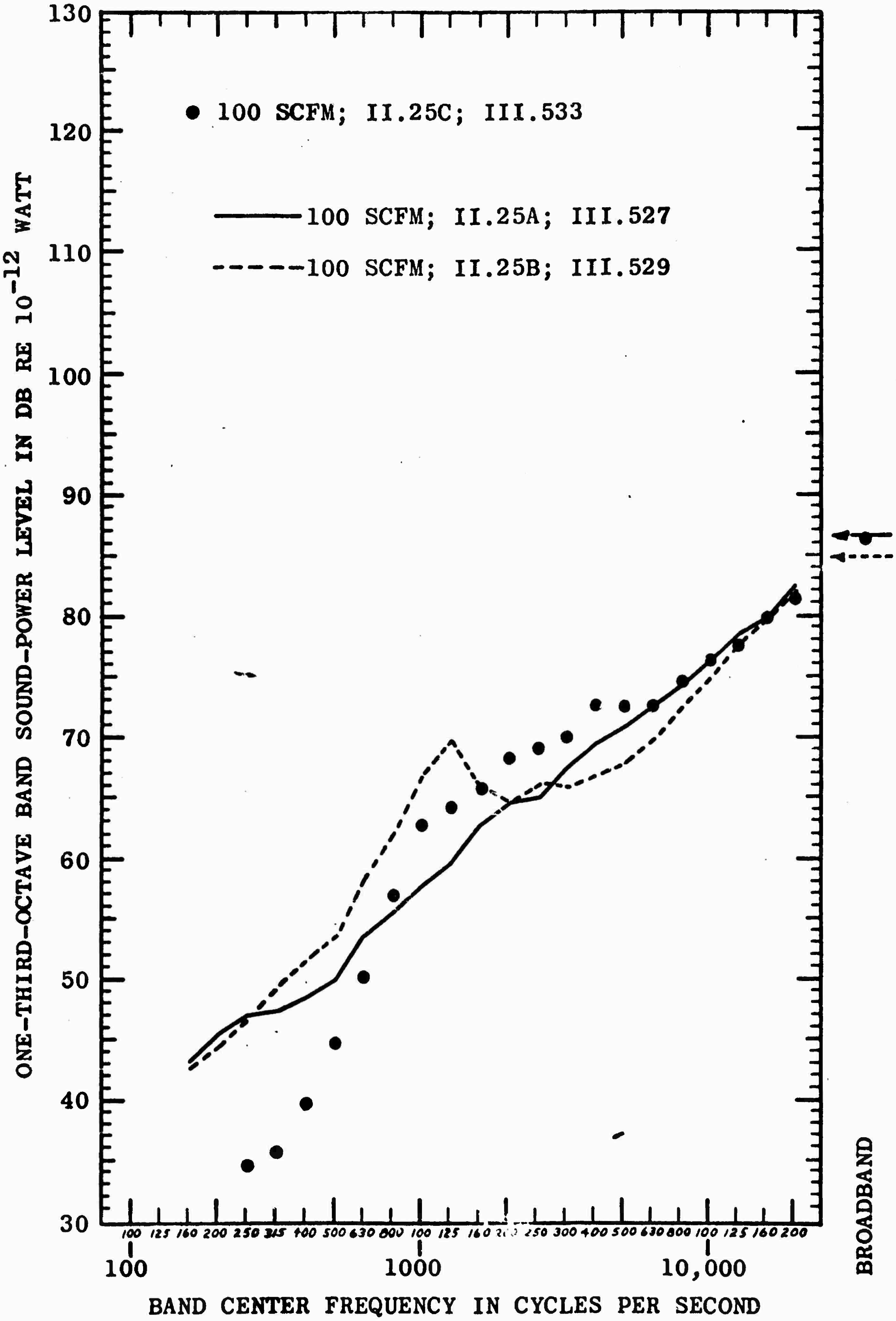

FIGURE 72. EFFECT OF TERMINAL LAYER OF METAL FELT.

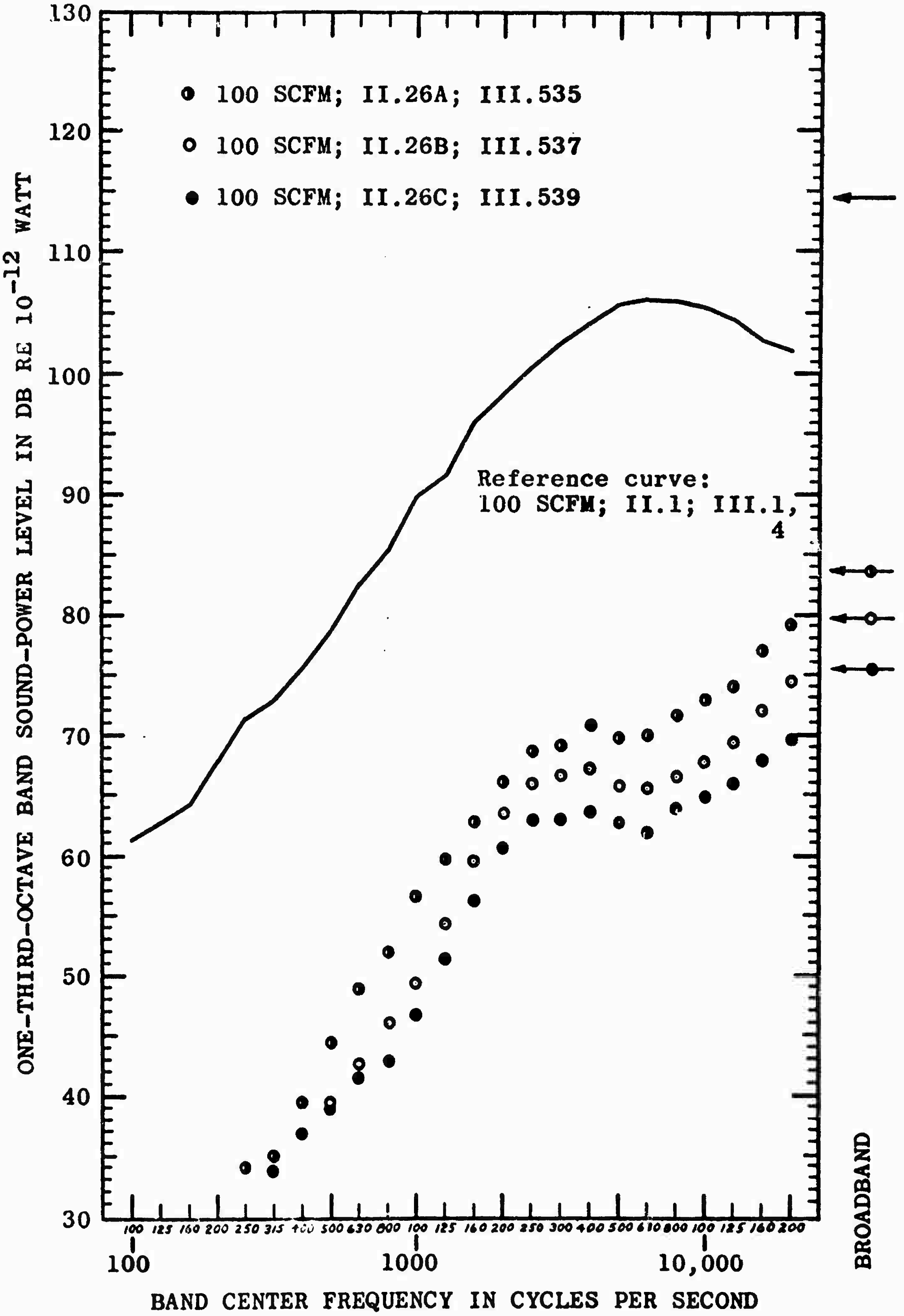

FIGURE 73. COMPOUNDED FELT-SCREEN SILENCERS; HIGH VELOCITY.

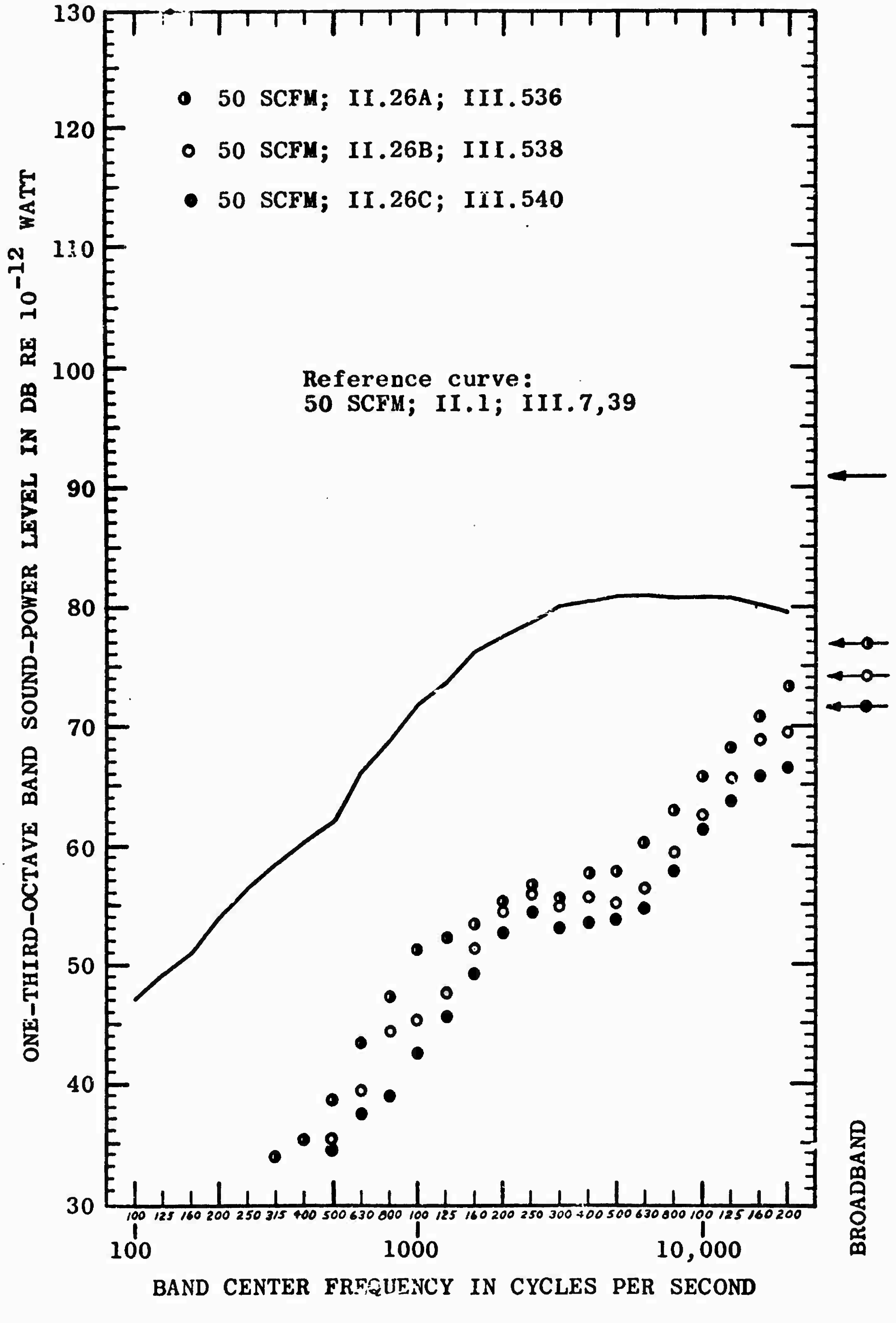

FIGURE 74. COMPOUNDED FELT-SCREEN SILENCERS; MEDIUM VELOCITY.

existing Air Force runup silencers[14]. However, no attempt was made to represent either the details or the internal structure of any existing silencers. This model-sized configuration of tubing is called a "muffler" body in this report.

The first set of experiments dealt with radiated sound power as a function of the location of the muffler body with respect to the 0.500 diameter nozzle extension. In the first instance, the body was placed tightly against the nozzle thereby precluding induced air flow at the nozzle end of the muffler body. In the other two cases, the body was located either even with the end of the nozzle extension at $x/d = 0$ or further downstream at $x/d = 2$ and induced air flow occurred. The noise spectra for the conditions $x/d = 0$ and $x/d = 2$ were so nearly identical that only the data corresponding to $x/d = 0$ are plotted on the following graphs. Figures 75 and 76 exhibit the experimental results for the high and medium velocity test conditions. Two major results stand out clearly. First, there was a pronounced increase in the sound power radiated, especially at low frequencies (At a linear scaling of 1:40 cps on these graphs would correspond to 10 cps for a full-sized jet.). Secondly, the presence of the muffler body contributes a distinct bumpiness to the spectra. Also, in some cases, a slight reduction occurred in the higher-frequency bands.

These findings are really to be expected on the basis of physical acoustics although the magnitudes would be difficult to predict from theoretical considerations. The frequency and spacing of the first several bumps in the spectra occur precisely at the normal mode frequencies expected for a pipe either open at both ends or open at one end and closed at the other end. The frequencies correspond to a pipe length of about 17" which closely approximates the center-line length of the muffler body. See Section 5.1. (More precise calculation is not warranted in view of the limited frequency resolution of the third-octave band data, some uncertainty about the magnitude of end corrections under finite flow conditions, and uncertainty about the acoustical "length" of a right-angled bend.)

Superimposed on the tube-like normal modes in Figures 75 and 76 is what appears to be an enormous increase in the broadband sound power radiated at low-frequencies. It is generally known that any solid surface in the vicinity of turbulence will enhance the acoustic radiation and the above results constitute another confirming example.

The small amount of reduction which sometimes appears at high frequencies could result from the interaction of several effects. The muffler body encloses that portion of the jet where the high frequencies are normally generated.

[14]Studies of this type were urged by Mr. Melvin Roquemore, U. S. Air Force Systems Engineering Group, to provide an obvious tie between this research on small models and some Air Force hardware for which acoustic data and operational experience were available within the Air Force.

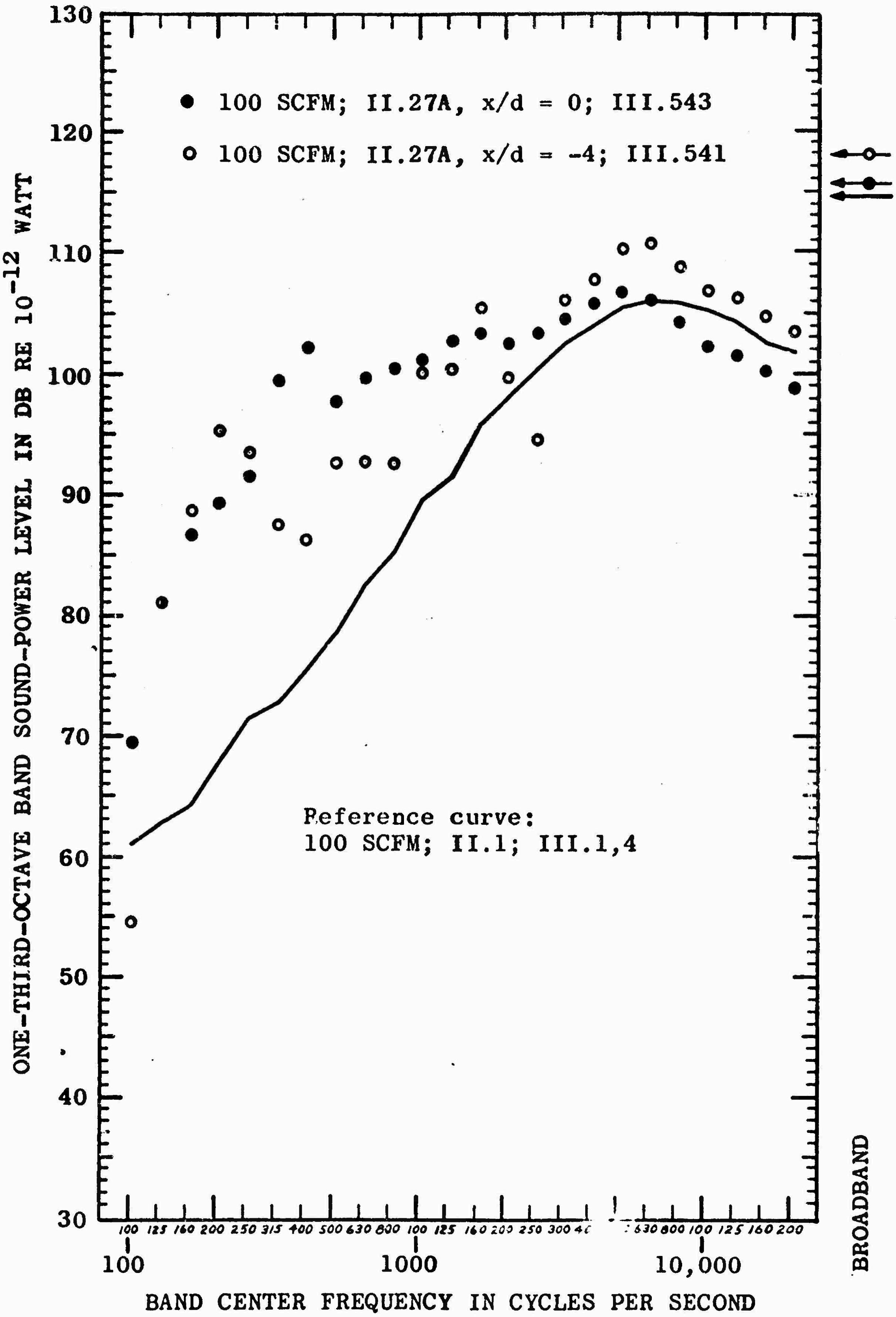

FIGURE 75. EFFECT OF BARE STEEL MUFFLER BODY; HIGH VELOCITY.

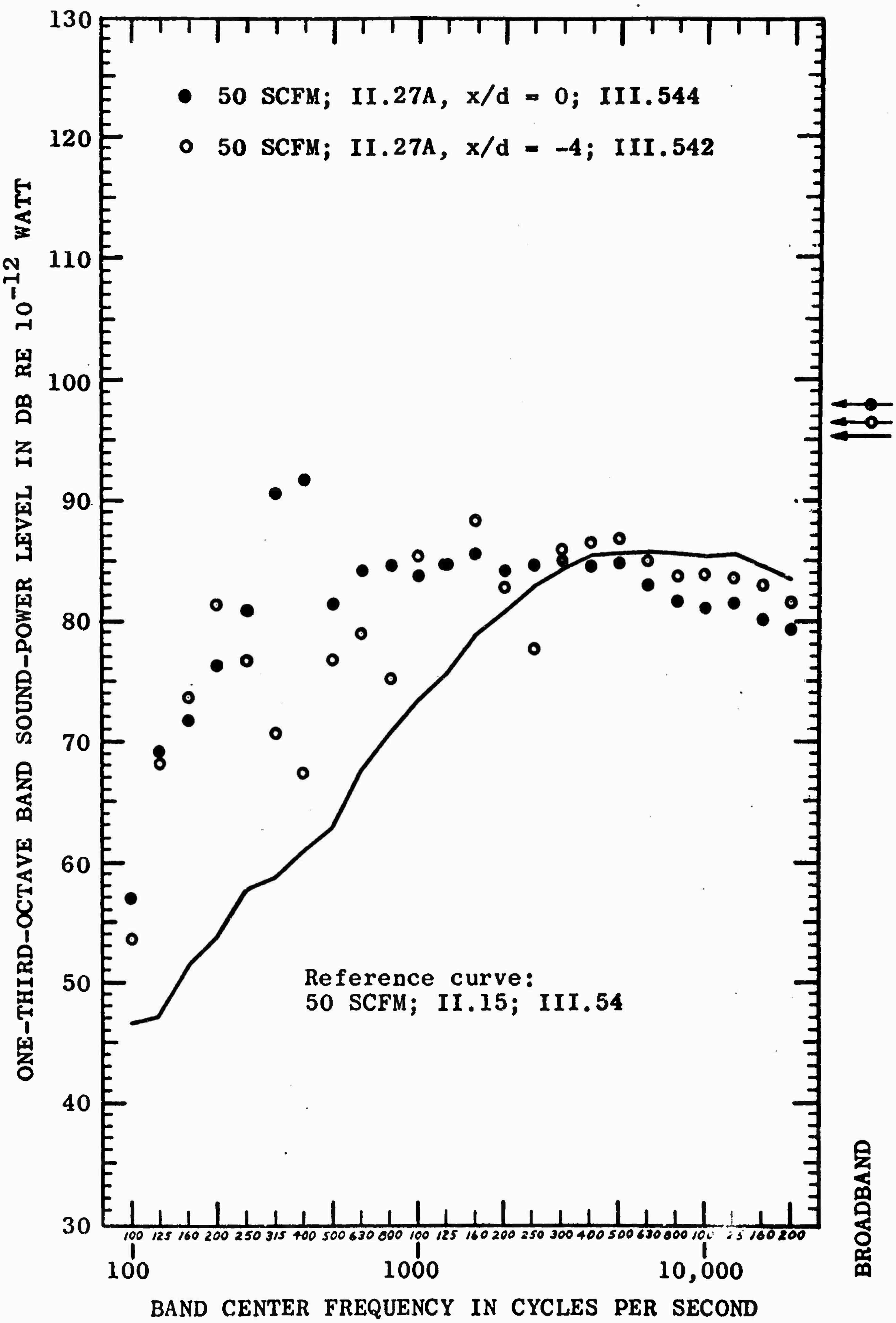

FIGURE 76. EFFECT OF BARE STEEL MUFFLER BODY; MEDIUM VELOCITY.

Consequently, this noise must propagate to the ends of the muffler body before it can escape and it seems to suffer attenuation in this process even though the steel muffler is essentially non-absorptive.

In order to varify that the results in Figures 75 and 76 really did stem from solid surfaces arranged in an acoustic pipe-like configuration, a similar muffler body was fabricated from perforated sheet metal and tested under similar conditions. The solid area of its walls was reduced to 65% by 1/8 inch diameter holes located in a triangular pattern, 0.185 inch on centers. Figures 77 and 78 show the results in the form of solid black dots. For comparison, the results with the solid muffler body are displayed with a short-dash line. In the case of the perforated muffler body, there was practically no acoustical difference whether the end of the body was placed tightly against the nozzle or placed even with the end of the nozzle extension. That is, the closed vs open pipe distinction has been eliminated by the perforations. It also follows that the spectra have become smooth in the absence of pronounced normal-mode behavior. However, the perforated muffler body contributes enhanced low-frequency noise although not to the extent caused by the solid muffler body.

In order to continue this line of investigation, the perforated muffler was wrapped on the outside with a single layer of fine fiberglass. The fiberglass had a very thin coating of neoprene on its outside surface which rendered it practically impervious to air flow. (This particular fiberglass was just a laboratory sample of uncertain commercial designation but definitely within the scope of commercial products.) Figures 79 and 80 illustrate the results which are plotted as solid black dots while the short-dash lines repeat the results for the perforated muffler body without the fiberglass wrapping. The effect of the absorption in reducing the high frequency noise is clearly displayed. Interestingly enough, the low-frequency noise is slightly increased over that for the bare perforated muffler body. It is as if the thin neoprene-coated fiberglass increased the amount of rigid wall experienced by the low-frequency noise. This increase in the low-frequency noise was small in magnitude but nevertheless real and consistent over a span of several octaves. Had the fiberglass been placed inside the solid-walled muffler body, one would anticipate, on the one hand, the increase in low-frequency noise and evidence of normal mode structure as illustrated in Figure 75 and, on the other hand, somewhat enhanced absorption due to the rigid backing of the fiberglass. Thus there would be competing effects with an indeterminate result; perhaps a little better or a little worse than in Figure 79.

It has been suggested in some of the open literature about runup silencers that resonant flexural vibration of the muffler body probably was responsible for excessive low-frequency near field noise. In the present research with models, various tests were made to find out if wall resonances contributed significantly to the radiated sound power. For example, the arrangement of the clamps, used for holding the muffler body in position, was altered in ways which would affect mechanical resonances but no change could be detected in the radiated power spectrum. Indeed, no flexural resonances of the walls

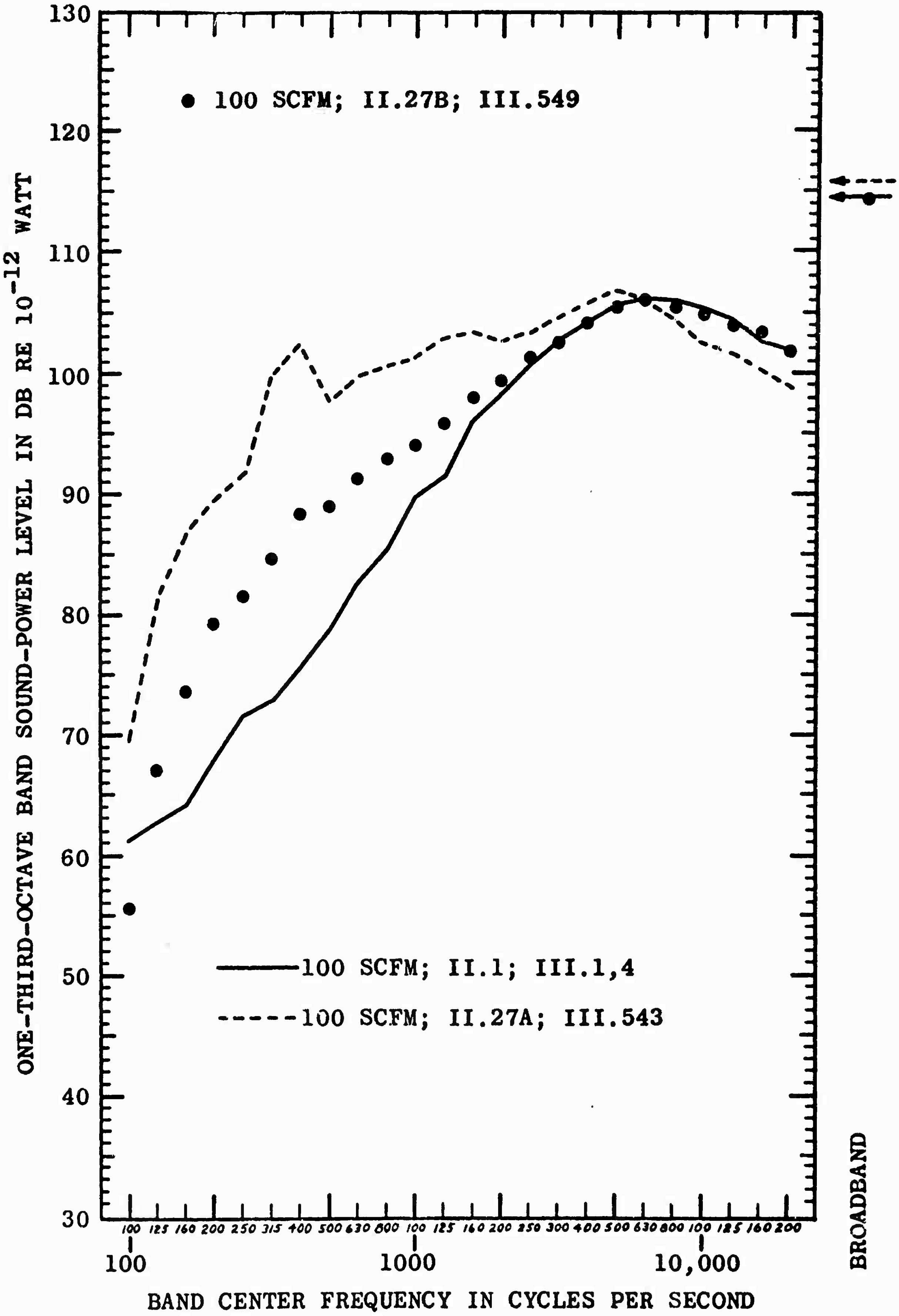

FIGURE 77. EFFECT OF PERFORATED MUFFLER BODY; HIGH VELOCITY.

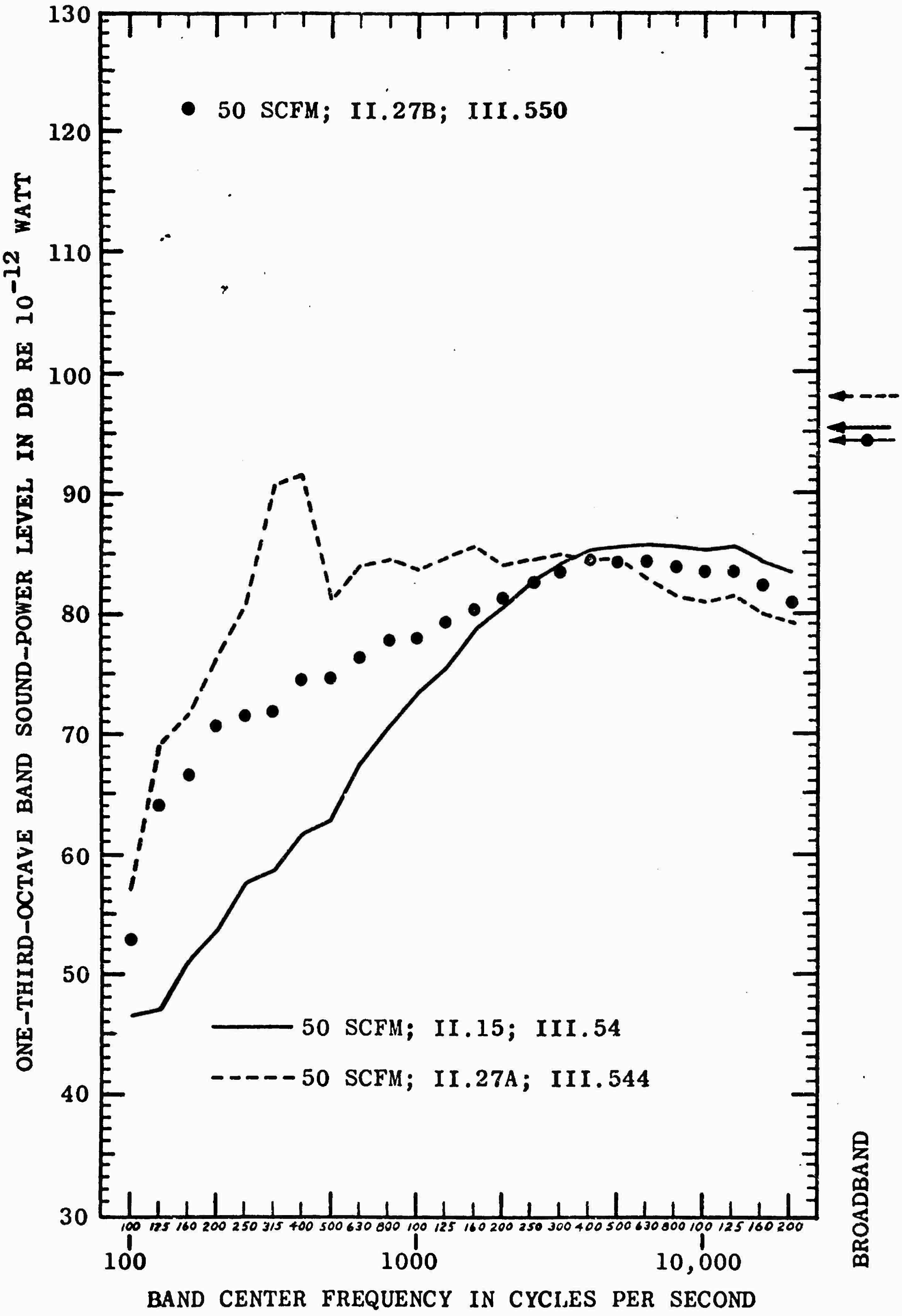

FIGURE 78. EFFECT OF PERFORATED MUFFLER BODY; MEDIUM VELOCITY.

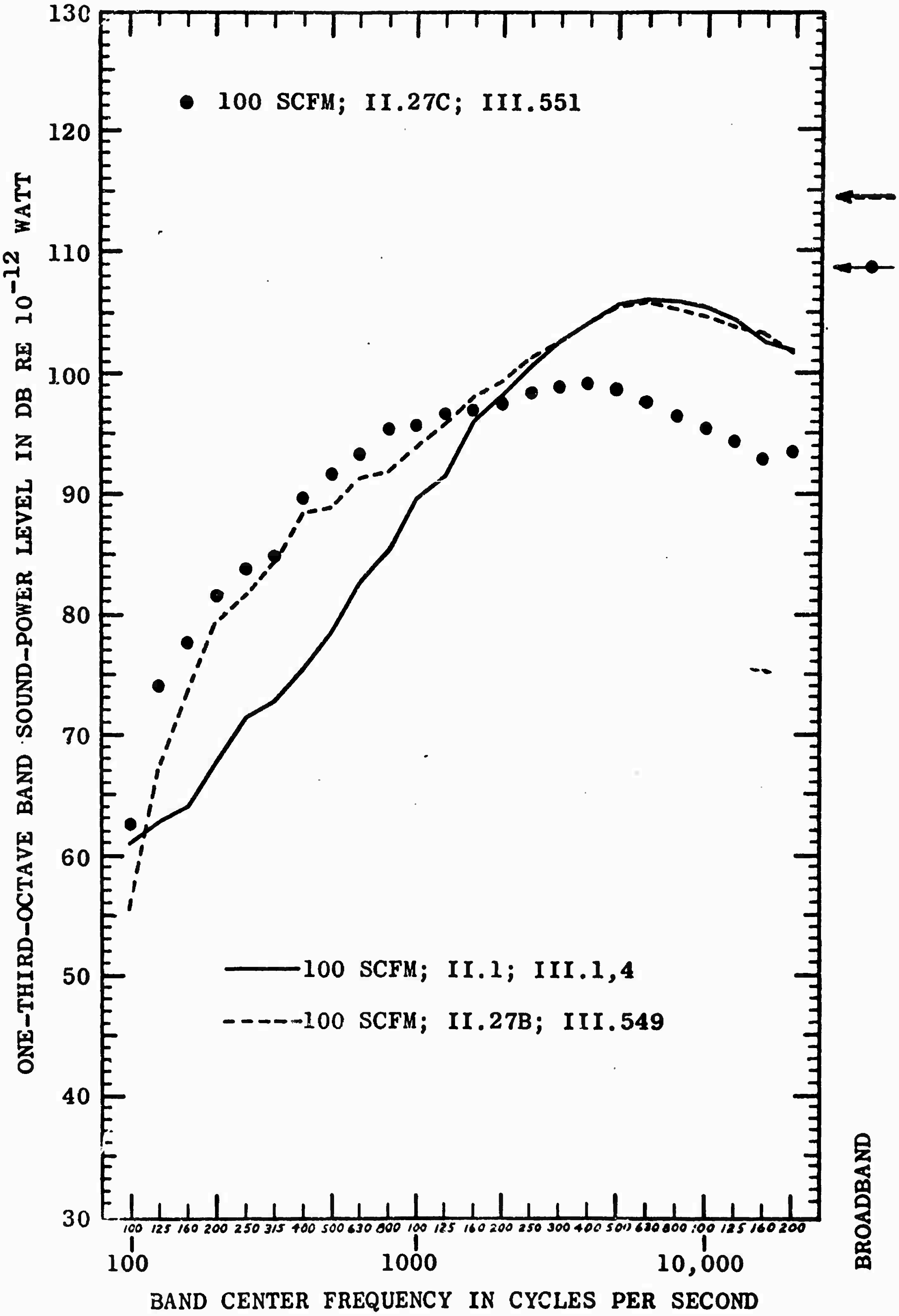

FIGURE 79. FIBERGLASS WRAPPED MUFFLER BODY; HIGH VELOCITY.

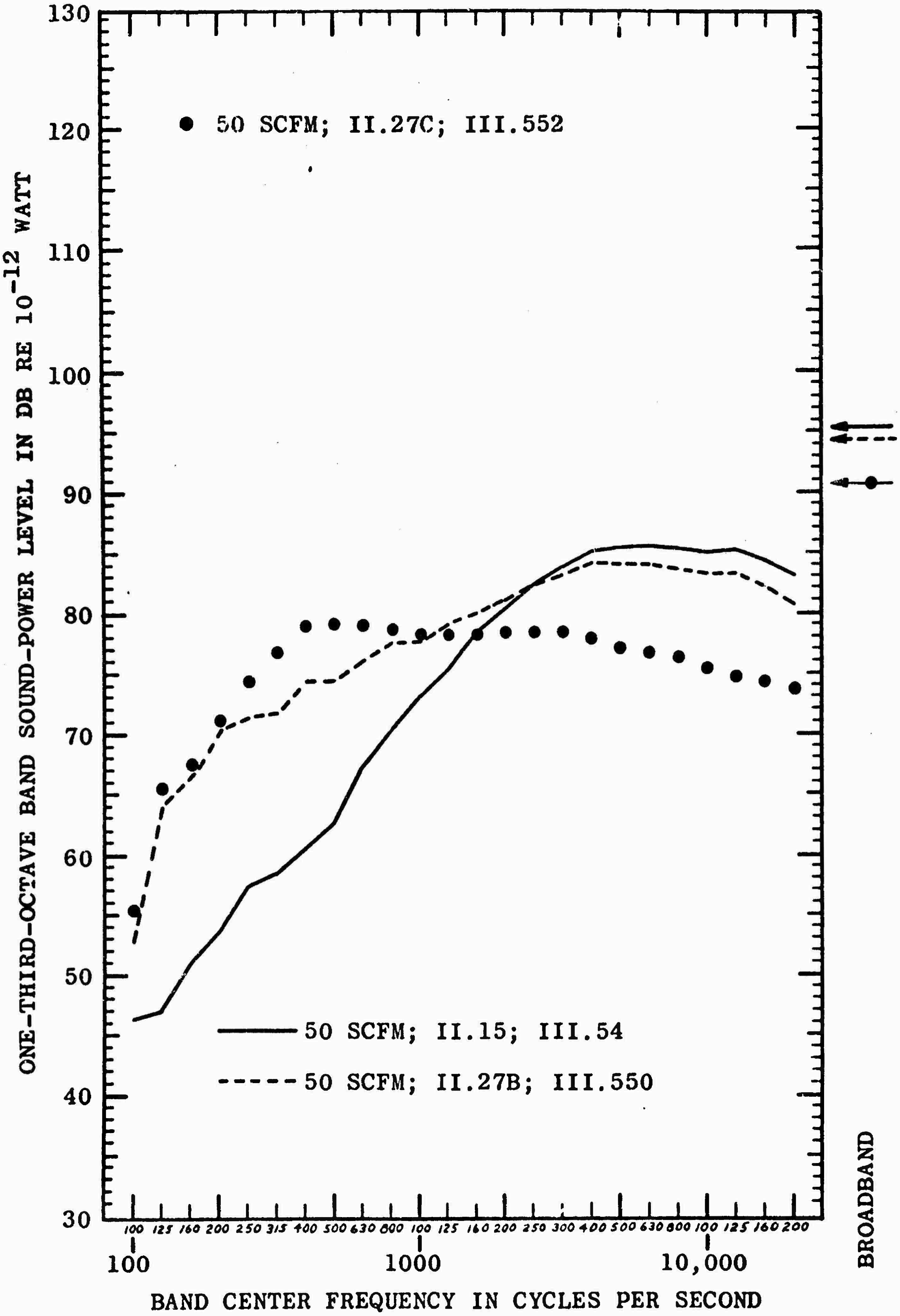

FIGURE 80. FIBERGLASS WRAPPED MUFFLER BODY; MEDIUM VELOCITY.

of the muffler body were ever detected during this research program. Moreover, the enhanced low-frequency noise occasioned by the muffler bodies was both at too low a frequency and distributed continuously over too wide a frequency range to be caused by mechanical resonances of the structures.

As a further check on the role played by the wall of the muffler body, it would be interesting to experiment with walls which would be impervious to air flow but which would be compliant in contrast to steel. At the instigation of Air Force Project Engineer, the Air Force supplied a model muffler body fabricated by the Goodyear Aerospace Corporation from their airmat material. This flexible model had approximately the same interior dimensions as the metal muffler bodies. It was a double-walled structure fabricated from a rubberized fabric and inflated with water. The total weight when inflated with water was about 17.5 kg. This model would seem to constitue an impervious, limp-walled, massive structure (however, the detailed impedance characteristics of its walls at audio frequencies are unknown) in contrast to the impervious, stiff, massive walls of the solid steel muffler body. The acoustical results obtained with this flexible muffler body are shown as solid dots in Figures 81 and 82. The short-dash lines represent the previous results for the solid steel muffler body. Clearly, the flexibility of the water-inflated structure provides no significant acoustical consequences. Peaks occur in the spectra at the locations expected for an acoustical pipe of these dimensions which is open at both ends. Moreover, the large general increase in low-frequency noise attributable to the presence of acoustically-opaque boundaries is as pronounced as for steel walls. At high frequencies, the water-filled model is not quite as noisy as the steel shell; probably due to slightly more acoustical absorption existing in its interior or to the higher transmission loss of its walls.

The results of these experiments with model muffler bodies appear to be completely consistent with the following interpretation.

A. Enhanced radiation will occur at the frequencies corresponding to the acoustical normal modes of the muffler body and are determined by body geometry and dimension. Furthermore, the mode frequencies are at or very close to the values expected for a zero dc flow condition.

B. There is appreciable enhancement of the radiation at low frequencies, apart from the normal mode frequencies, which is due to the presence of acoustically opaque boundaries in the vicinity of the jet's turbulent noise sources. The magnitude of the enhancement seems to be of the order of 10 to 20 db.

C. Flexural resonances and coincidence effects do not appear to have played any significant role in these test results. If such phenomena did occur, they could only worsen the acoustical situation by resulting in even more radiation.

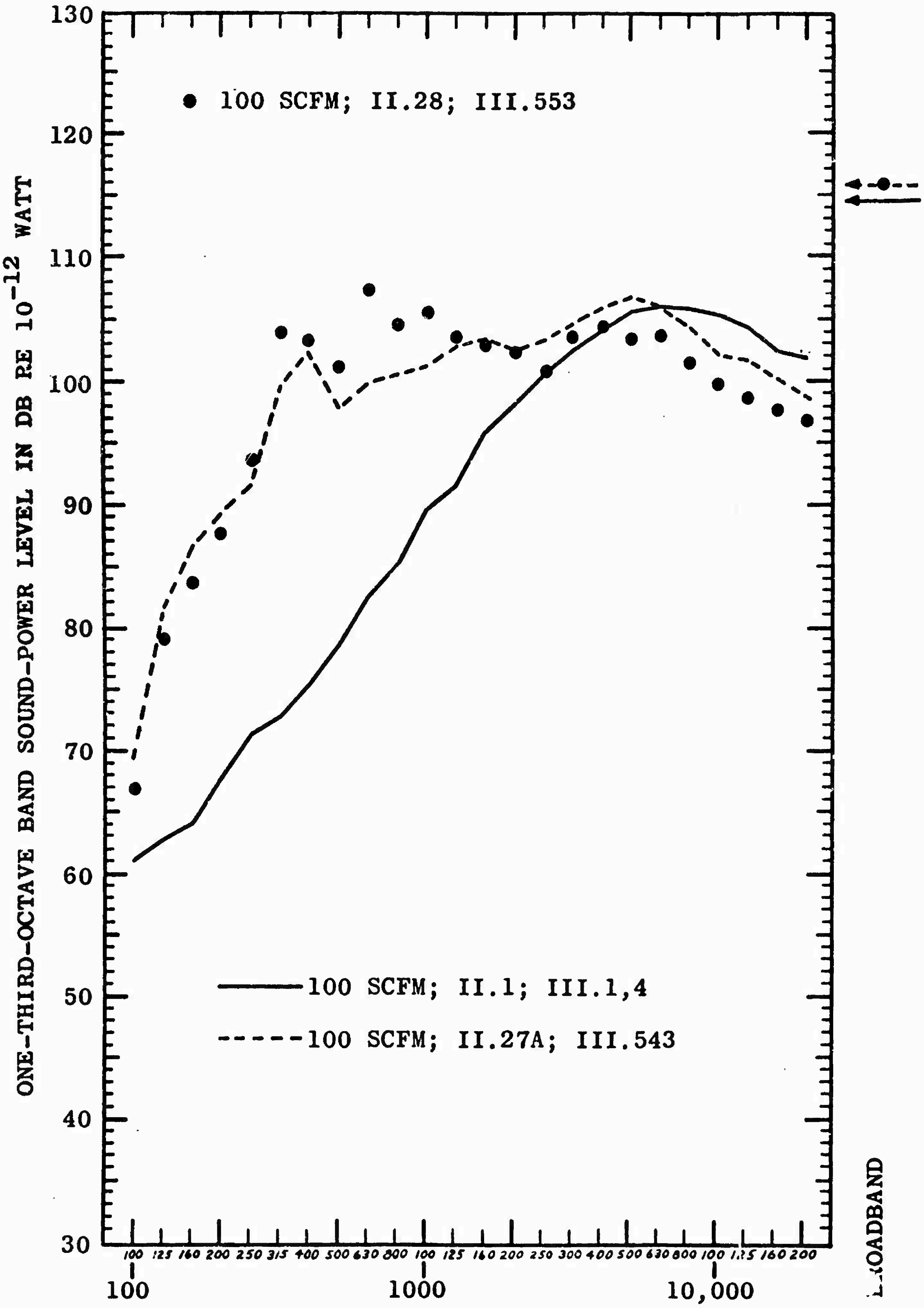

FIGURE 81. WATER-INFLATED FLEXIBLE BODY; HIGH VELOCITY.

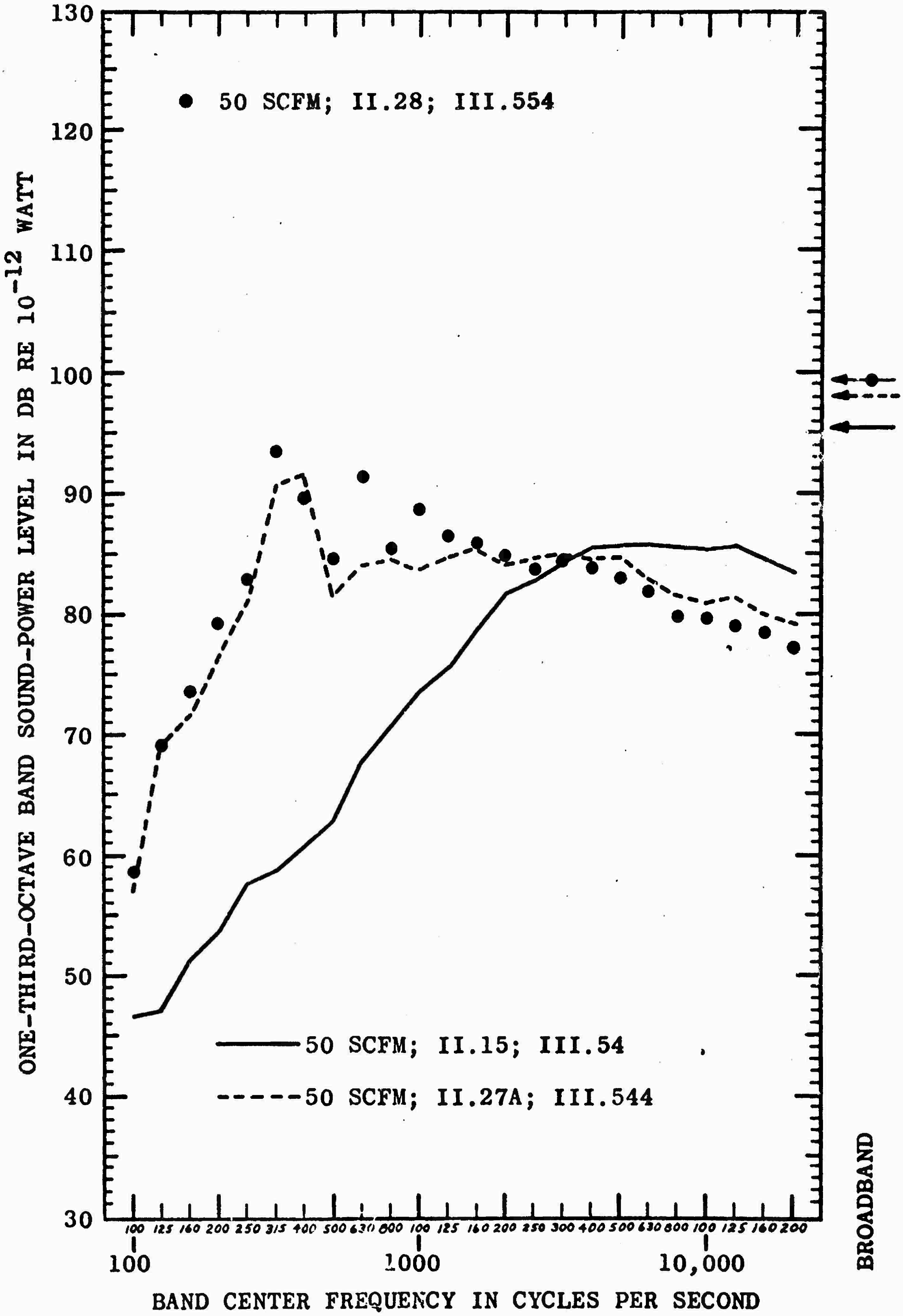

FIGURE 82. WATER-INFLATED FLEXIBLE BODY; MEDIUM VELOCITY.

D. In the absence of a specific noise reduction mechanism, such as absorption, little or no silencing can be expected. (The small reductions observed at high-frequencies are due perhaps to partial shielding of the sources in such a way as to take more advantage of the natural absorption of the atmosphere.)

The above conclusions are not at all unusual or unexpected; indeed, they are precisely what should be expected from our current knowledge of acoustics. They should apply to full-sized mufflers with equal validity as long as the muffler dimension to wavelength ratio is preserved.

We did not extend this line of investigation to include muffler bodies lined with absorption. To do so would have paralleled the development of some of the existing full-sized runup silencers and our research contract specifically admonished against such a course of research. Technically, investigation of absorptive interior treatments is of very limited value on a model scale beyond the elementary demonstration that absorption is a potentially useful mechanism. Serious problems generally accompany attempts to scale absorption over large frequency ranges.

6.4. MUFFLER BODIES WITH SCREENS AND METAL FELTS

Several experiments were conducted with the muffler bodies used in combination with screens and metal felts. We did not expect any of these experiments to yield a superior muffler but rather we wanted to find out if the acoustical results would be additive or whether more complicated interactions might occur. As already mentioned, research with absorptive linings was deliberately omitted.

In one type of experiment, a screen was placed across the entrance end of the muffler body and the combination mounted to locate the screen at $x/d = 1/2$ with respect to the nozzle extension. Figure 83 and 84 summarize the results, shown as solid black dots, for the 20 mesh screen and the solid muffler body. For comparison, the open circles represent the solid muffler body alone while the short-dash lines represent the results for the 20 mesh screen alone at $x/d = 1/2$. By inspection it is clear that the muffler body and the screen act practically independent of one another and that the spectrum for the composite can be obtained to a reasonable approximation by subtracting the silencing previously found for the screen alone from the spectrum for the muffler body alone. The evidence of normal mode behavior for the muffler body persists as would be expected.

When screens of finer mesh are combined with a muffler body, the situation appears to become more complicated. Figure 85 shows some results of combining the 60 mesh, 16% open area screen with the solid muffler body. The solid black dots represent the condition when the screen was placed across the exit of the muffler body. Compared with the short-dash curve which re-

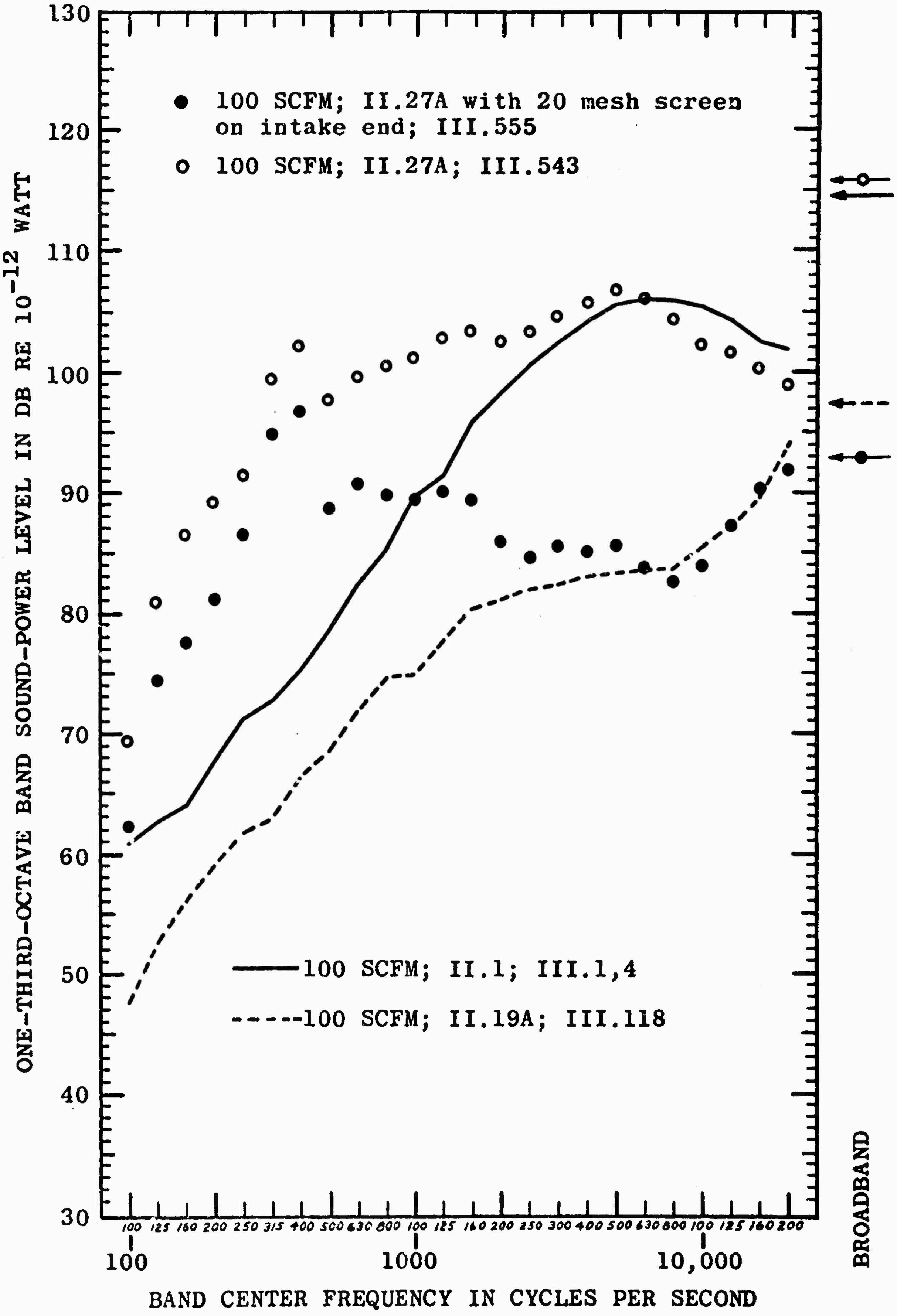

FIGURE 83. SOLID MUFFLER WITH 20 MESH SCREEN; HIGH VELOCITY.

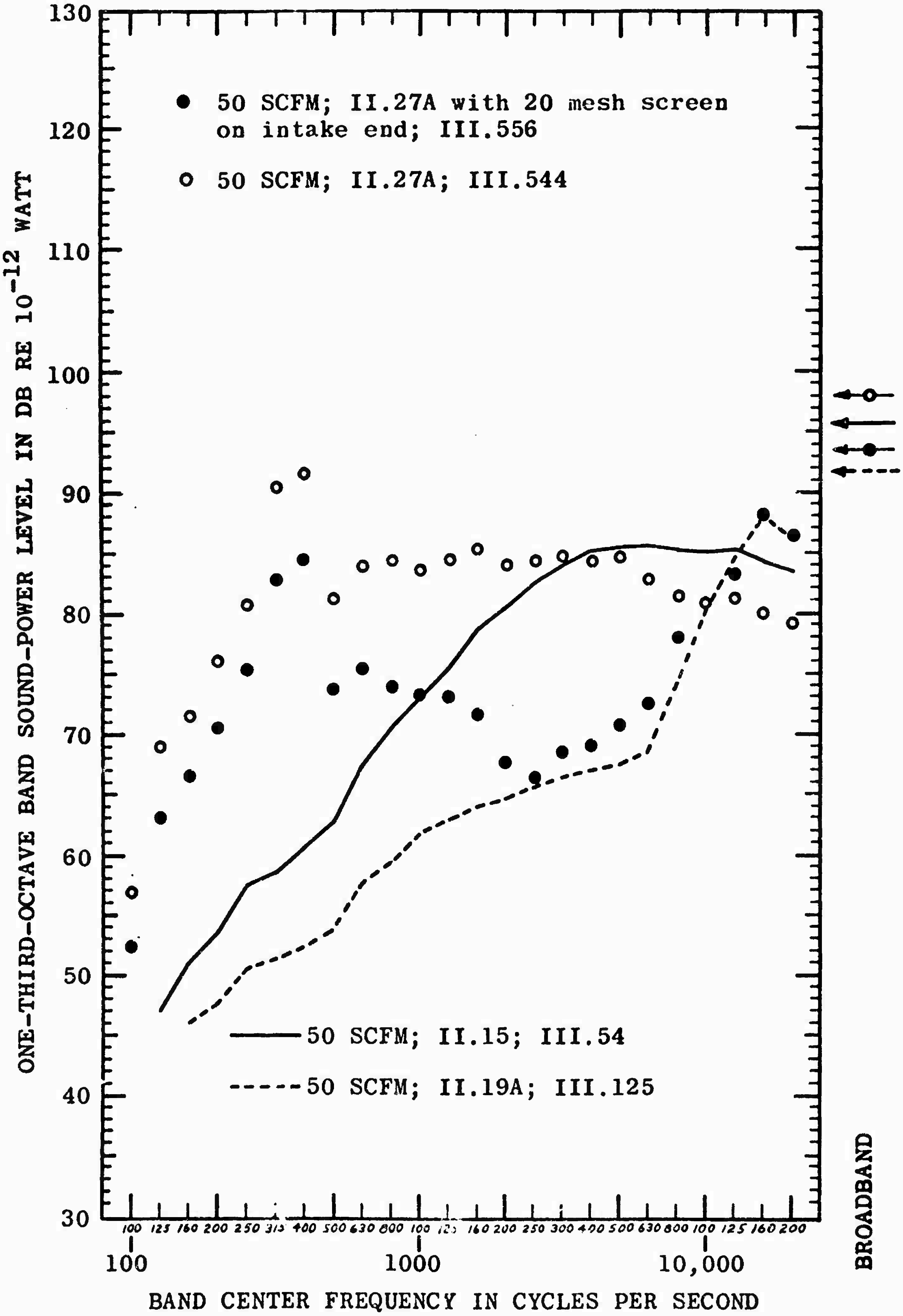

FIGURE 84. SOLID MUFFLER WITH 20 MESH SCREEN; MEDIUM VELOCITY.

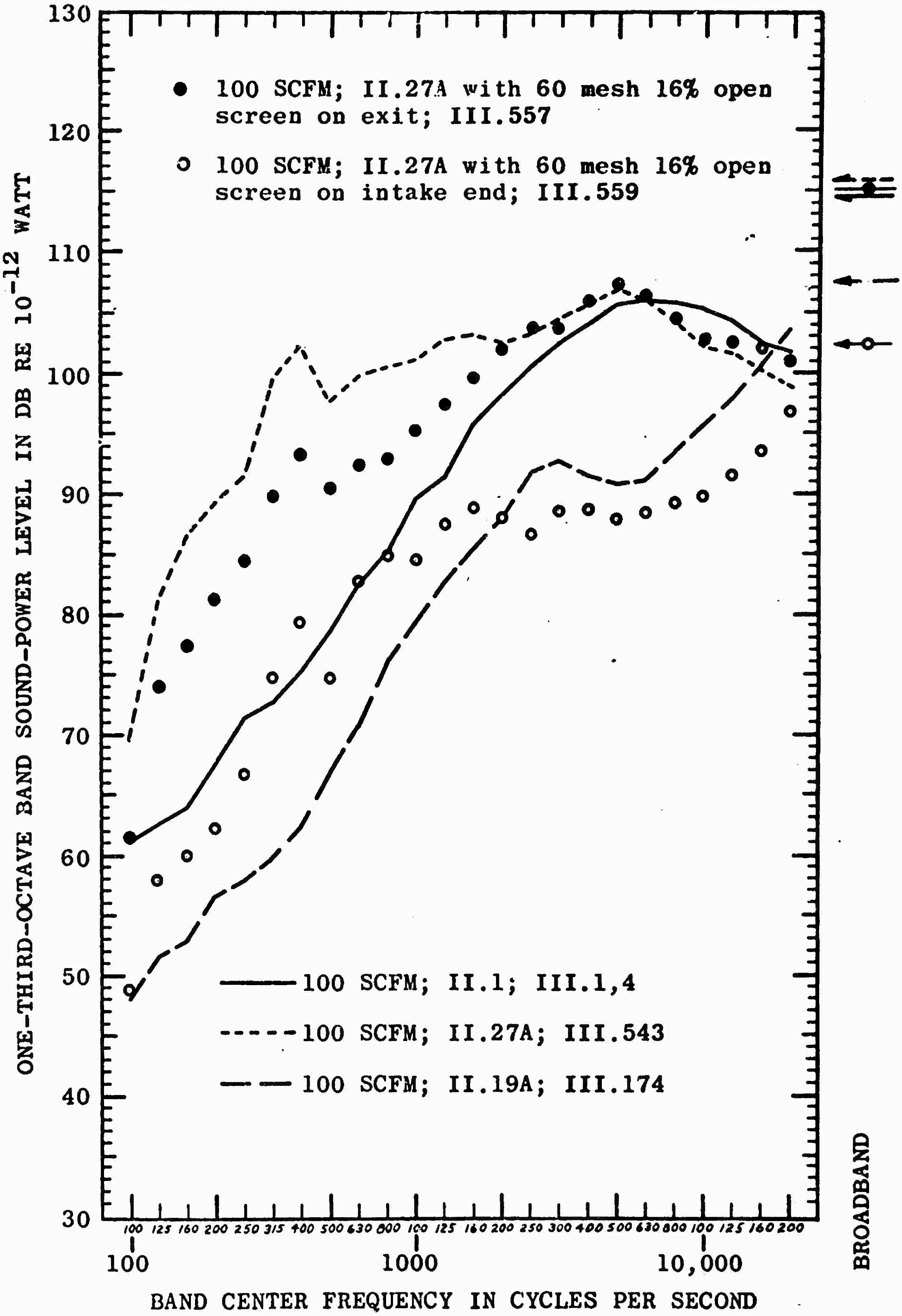

FIGURE 85. SOLID MUFFLER WITH 60 MESH SCREEN; HIGH VELOCITY.

presents the muffler body alone, a silencing action is evident below 2000 cps but even so, the noise remains greater than for a bare jet.

When the 60 mesh, 16% open screen was placed on the intake end of the muffler body and located at $x/d = 1/2$ (see the open circles in Figure 85) appreciable silencing occurred across the whole spectrum. At low-frequencies, the band levels approached those for the unsilenced jet while above approximately 1000 cps useful amounts of silencing are evident.

The long-dash line in Figure 85 represents the results for the 60 mesh, 16% open screen tested alone at $x/d = 1/2$. In the present experiment, the effect of the screen located on the intake end of the muffler body appears to be much larger than for the same screen used alone. The high reading in the 400 cps band varifies that the pipe-mode behavior has not been drastically altered by the partial closing of one end of the pipe with a screen. At high frequencies, this arrangment might constitute a frequency filter describable by acoustical circuit theory but the breadth of the effects, spanning almost seven octaves, tends to refute such an explanation.

When the 60 mesh 37% open screen was placed on the exhaust end of the muffler body, a spectrum very similar to that for the 60 mesh 16% open screen (solid black dots in Figure 85) was obtained except that it was two or three db noisier below 2000 cps. However when this more open 60 mesh screen was placed on the intake end of the muffler body, a very pronounced silencing effect occurred between about 500 and 10,000 cps. This result is demonstrated in Figure 86 while Figure 87 demonstrates that a similar behavior is to be found at lower jet velocities also. At the pipe mode frequency near 400 cps and at lower frequencies, no reduction in noise below that for the unsilenced jet is to be found. However, in this frequency range, we are well below the peak spectral levels and so reduction of the noise in this portion of the spectrum may not be essential for some applications. Figures 86 and 87 show rather large residual amounts of noise at high frequencies but these frequencies, scaled down for full-sized jets, lie in a range where ordinary acoustical absorbing materials are most effective.

Results as extreme as those demonstrated in Figures 86 and 87 occurred only for the 60 mesh 37% open screen located on the intake end of the solid muffler body. Other screens and configurations may have exhibited vestiges of such effects but if so, they were too insignificant to be recognized in this first study of the reduced data. There was no research time left in which to investigate the nature of the effects displayed in Figures 86 and 87 further. One might intuit an acoustical bandpass filter effect superimposed on the effect of the screen alone but a definitive interpretation would require considerable additional research. These results do indicate a possibly useful effect if it should turn out to be an effect which is not currently exploited in conventional runup silencers.

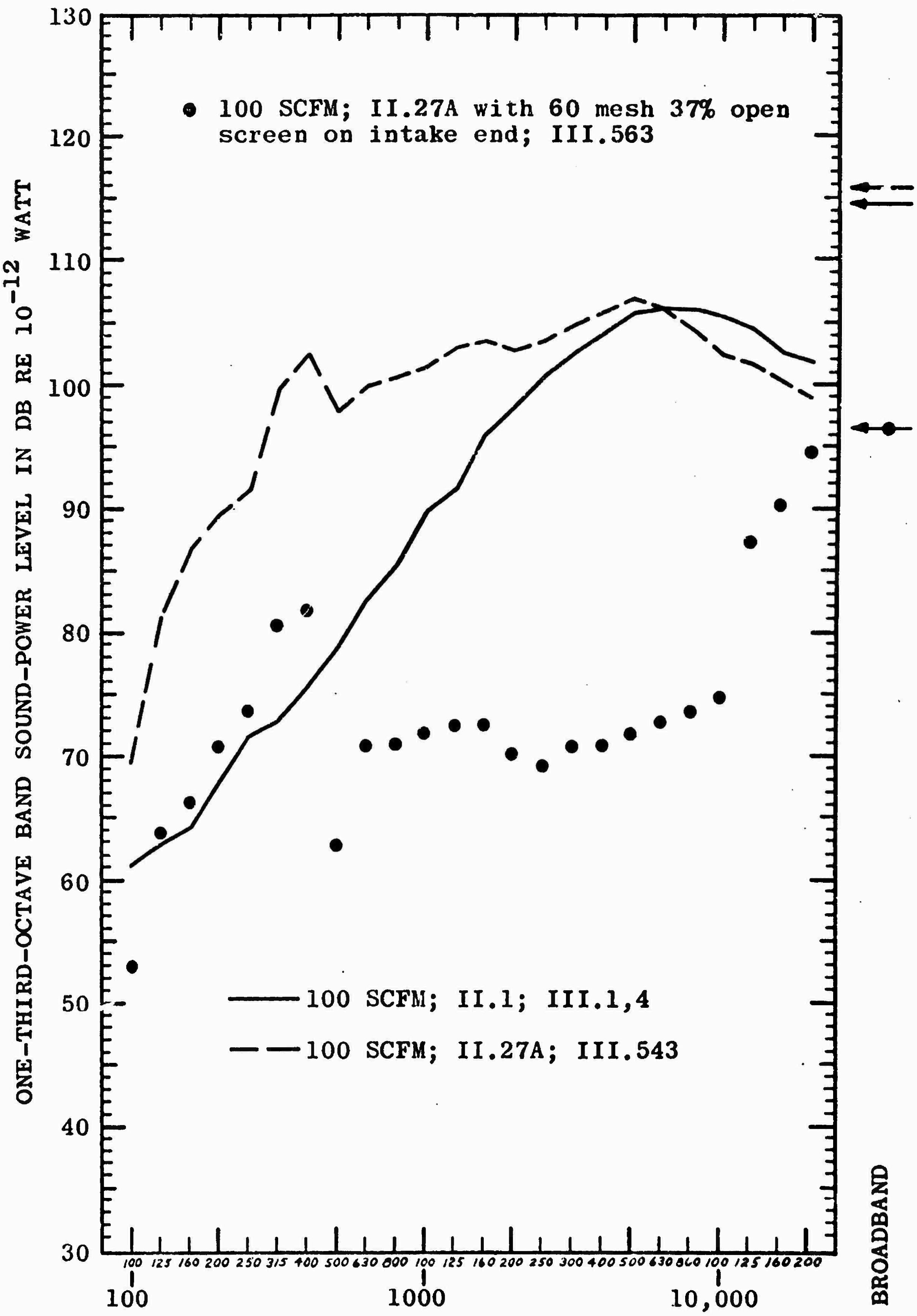

FIGURE 86. SOLID MUFFLER WITH INTAKE SCREEN; HIGH VELOCITY.

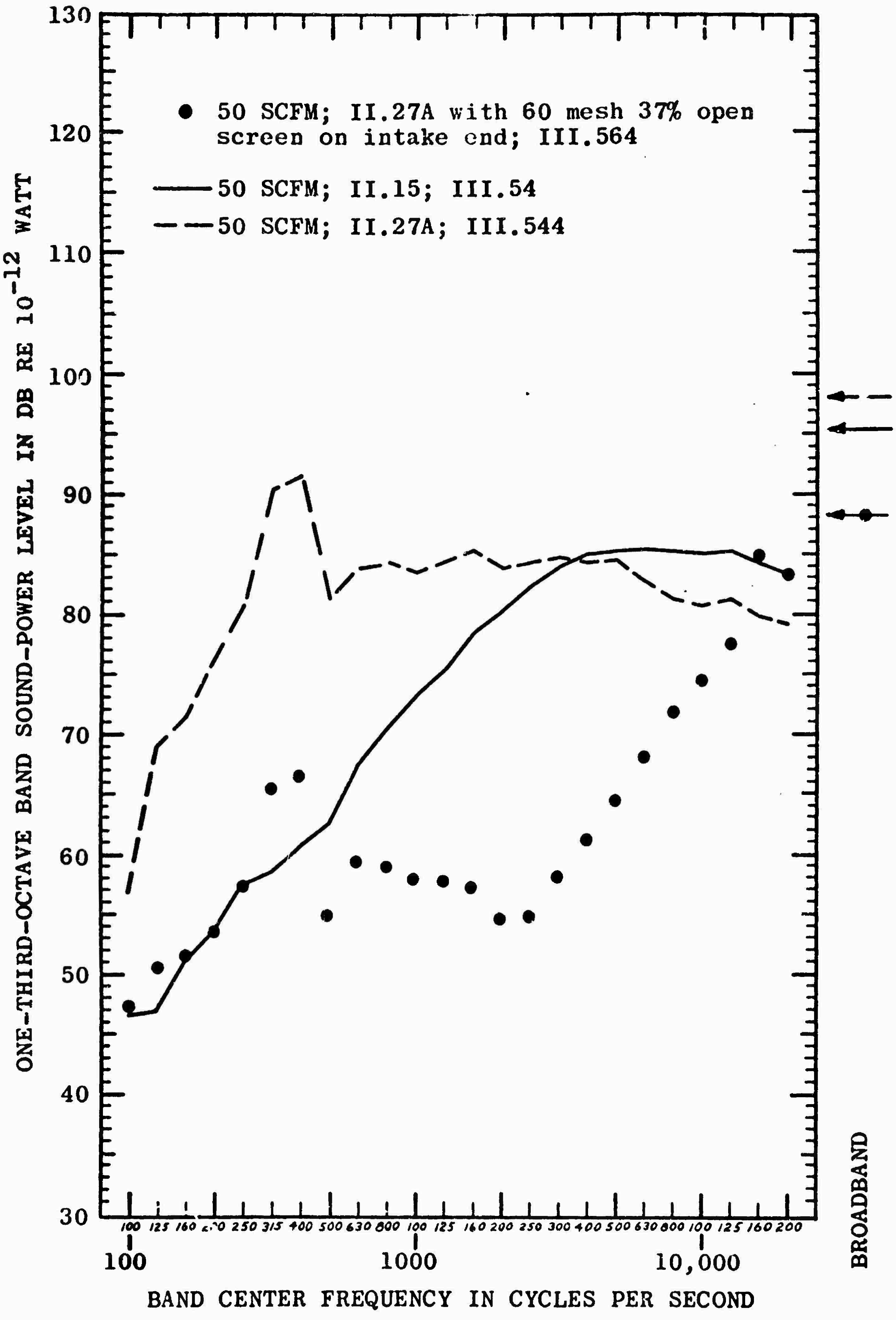

FIGURE 87. SOLID MUFFLER WITH INTAKE SCREEN; MEDIUM VELOCITY.

In some further experiments with combinations of objects, screens were placed on both ends of the solid muffler body. The terminal screen appears to have disrupted the acoustical conditions which led to the results shown in Figures 86 and 87 and instead gives results similar to the open circles shown in Figure 85. The results for the several tests of this type are not sufficiently interesting to warrent separate graphs.

In another group of experiments investigating interactions, a 1/8" thick layer of 5% dense metal felt was supported with the 60 mesh 37% open screen and this combination was located at $x/d = 1/2$ and placed on the entrance end of the solid muffler body while a 60 mesh 16% open screen was placed across the terminal end of the muffler body. Figures 88 and 89 shown the results. In these cases, the high frequency results do not appear drastically different from those for the metal felt alone or a metal felt supported by a screen. At low frequencies, evidence of the normal mode behavior for a pipe, open at both ends, appears. In these experiments, the results seems to be approximately an algebraic summation of the increases due to the solid muffler body and decreases due to a metal felt. Again, in these experiments and in the absence of an interior absorptive treatment, a muffler body seems to constitute an acoustical liability.

6.5. FINE DENSE SHOT INTRODUCED INTO JET EXHAUST

Several investigators have reported on the possible usefulness as a silencing mechanism of injecting water into the exhaust jet (see, for example, Reference 25). Water has also been used in some designs of runup silencers principally for cooling of the structure. The use of water for cooling of the silencer is inconsequential to the present investigation. In some designs, however, the water mixed with the hot jet exhaust and was discharged along with the exhaust gases as steam or perhaps as a fine mist under some operating conditions. Under such conditions, the water might very well affect the acoustical results and hence research directed toward understanding the possible acoustical mechanism would be appropriate. The open literature available during the present research program did not appear to provide a complete story. (The use of water in a runup silencer presents logistic problems but these are quite independent of whether water can physically provide a silencing action.)

From one point of view, the water would alter the composition and the thermodynamic state of the jet exhaust and as a consequence might alter the details of the noise generation processes. From another viewpoint, the water constitutes additional mass which upon being injected into the exhaust stream must be dispersed into droplets and accelerated to the velocity of the exhaust stream. The energy needed to disperse and accelerate the water must come from the jet hence the mean exhaust velocity would decrease and some noise reduction might accrue. The effects of the additional water on sound propagation in the surrounding atmosphere are probably too small, or even in the wrong direction, to account for useful amounts of silencing. The ques-

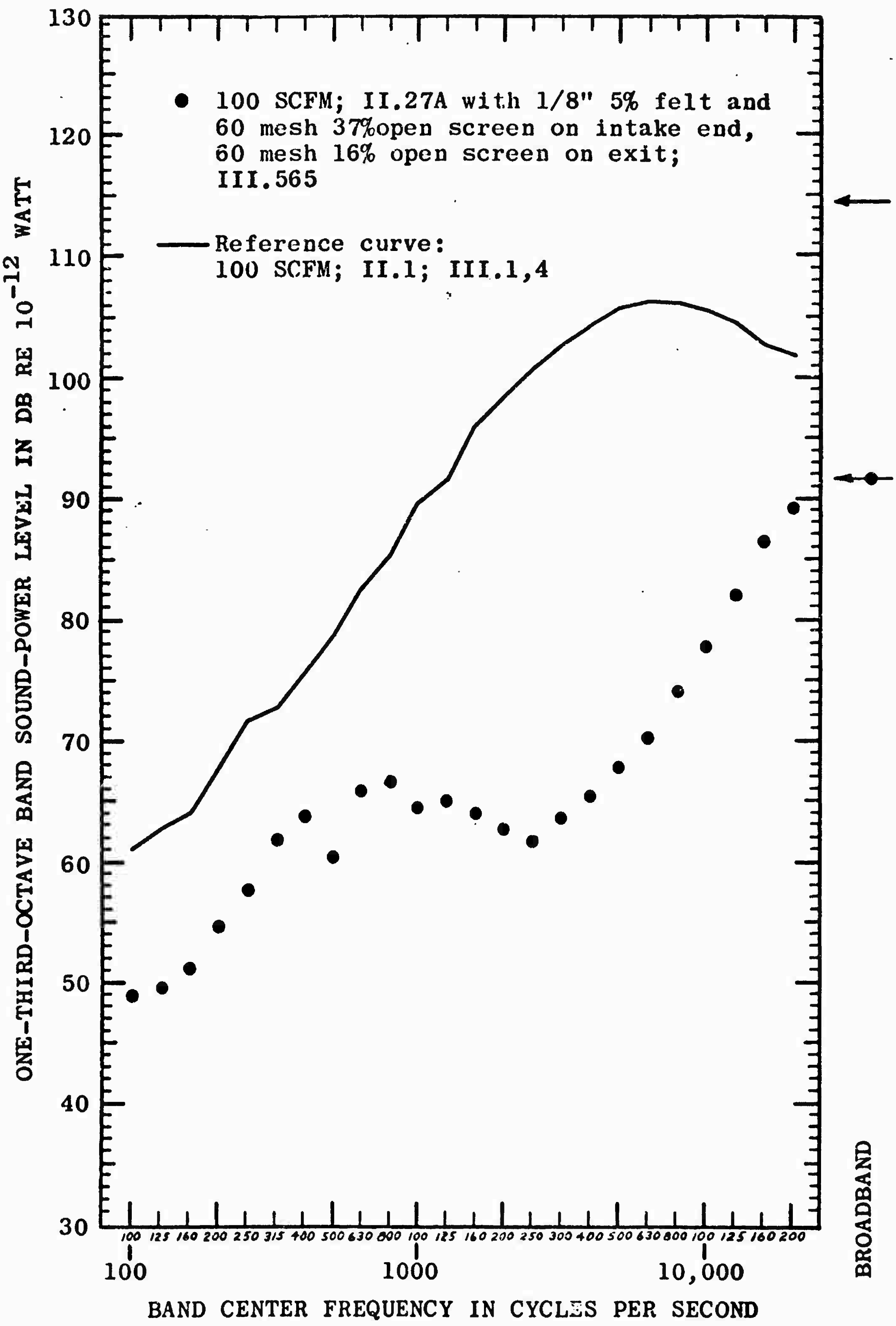

FIGURE 88. SOLID MUFFLER WITH METAL FELT; HIGH VELOCITY.

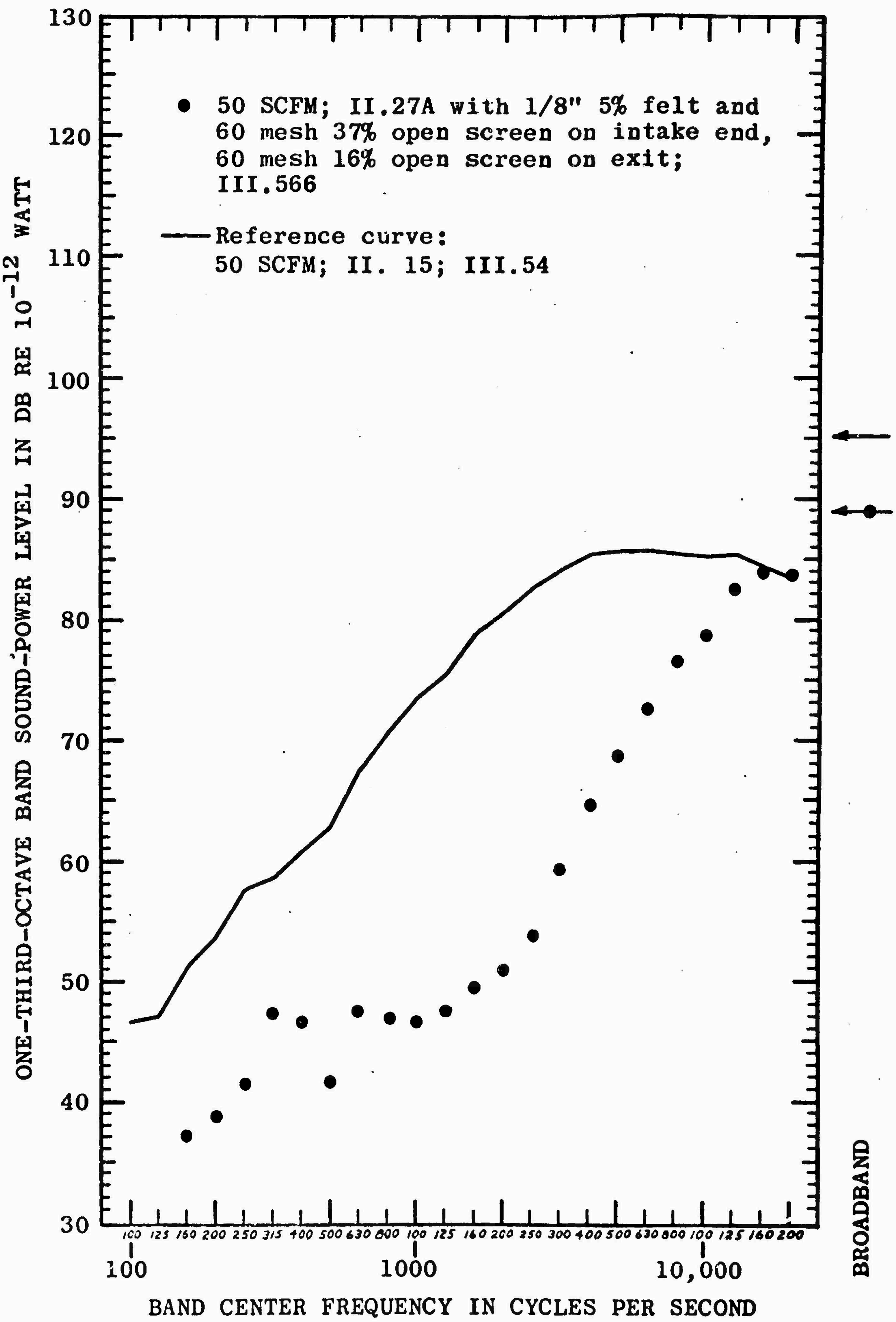

FIGURE 89. SOLID MUFFLER WITH METAL FELT; MEDIUM VELOCITY.

tion of water as a silencing additive would appear in the main to involve either thermodynamic or mechanical principles.

Our model studies with cold air jets did not lend themselves to investigation of thermodynamic consequences. Mechanical effects could be investigated in principle with cold air jets but the introduction of a water spray into the reverberation room would create troublesome measurement problems. The moisture content of the atmosphere within the reverberation room should remain practically constant during a test run to permit evaluation of the acoustical data in terms of sound power. Furthermore, condenser microphones and their preamplifiers tend to malfunction when exposed to high moisture conditions. However, one mechanical aspect, that of accelerating the heavy droplets, seemed capable of investigation by using other types of dense particulate matter.

We elected to try an experiment of this type using small spherical glass beads. The presence of glass beads in the reverberation room would not appreciably alter the prerequisite acoustical conditions for sound power measurement. Also, the glass beads would not dust but simply fall to the floor where they could be recovered with a vacuum cleaner.

To conduct this experiment, a large plastic funnel was located close to and slightly above the nozzle exit so that gravity would cause a stream of beads, nearly the same diameter as the nozzle, to fall into jet stream. The air stream would act to accelerate the beads at almost right angles to their original trajectories. Masking tape was used to seal the end of the funnel until the start of the test run when the tape was simply jerked away.

The supply of glass beads loaded into the funnel would last for only about 20 seconds, consequently steady-state conditions could not be maintained long enough to record a complete spectrum. Instead, two test runs were made, first observing the wide band noise and then observing the noise in the one-third octave band centered at 10,000 cps.

The glass beads used in these exploritory experiments were about 0.232 to 0.0116 inch diameter and had a density less than 2.99 according to the manufacturer's data.[15] A comparison of the mass rate of flow of the glass beads to the mass rate of air flow from the nozzle leads to a ratio of about 2 (the experimental data actually yields a value of 2.04). The acoustical consequences of the experiment were very small. The broadband measurements showed, at most, a 1.5 db reduction in the noise while the measurement of the 10,000 cps band showed, at most, a 1.5 db increase in noise. Considering the somewhat reduced accuracy of measurement for these tests, the above results should be interpreted as meaning no acoustical effect of consequence.

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[15]Type MS-XP Glas-shot supplied by Micro Beads Div., Cataphote Corporation, P.O. Box 28, Sta. F, Toledo, Ohio 43610.

On the basis of experimental data quoted in the literature for some tests with rocket engines (see Reference 25) one would have anticipated a reduction of 8 to 10 db at a mass ratio of two on the assumption that these diverse experiments were acoustically comparable. This brief exploratory experiment with glass beads is far from definitive, but in the absence of other more complete and reliable information, the essentially negative result suggests that explorations in other directions might be more fruitful for explaining acoustical effects of water injection.

SECTION 7

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APPENDIX I

EXPERIMENTAL FACILITIES

The research reported here was designed to take advantage of the reverberation room method for measuring sound power. The scale of the experiments was adjusted according to the size and characteristics of the available reverberation room facilities. The reverberation room proper has a rectangular shape and its internal dimensions are 22 feet long by 18 feet wide by 13 feet 6 inches high, yielding a volume of 5,346 cubic feet. The proper space- and time-average of the sound pressure field within the room is obtained through the use of a reflector vane, about 135 square feet in area, which rotates at about 12 rpm. This rotating vane effectively introduces a time-varying boundary condition which makes possible valid sound-power measurements even on sources which radiate discrete frequencies or narrow bands of noise. The sound power spectrum of simple jets, because of its known smooth envelope and continuous distribution with respect to frequency, could have been measured adequately in a reverberation room possessing only stationary boundary conditions. However, there could be no such a priori knowledge about the nature of the spectra to be produced by the various experimental configurations studied and consequently the rotating vane constituted the essential factor in assuring valid acoustical measurements. (There are several methods for obtaining the necessary average sound-pressure measurements in reverberation rooms, e.g., a diagonal traverse with the microphone, but so far, the rotating-vane method appears most practical, especially when narrow-band decay-rate measurements are also needed for the computation of sound power.)

The reverberation-room method for the measurement of sound power is particularly valuable because, when properly applied, it becomes an absolute method. That is, this method does not depend upon the use of secondary acoustical standards. (See References, 2, 9, 26 and 27) Indeed, the microphones employed can be directly calibrated by a diffuse-field reciprocity method so that ultimately the entire sound-power measurement can be conducted so that it rests upon one calibrated electrical meter of say 0.25% accuracy and the ratios of some precision resistors. And there is no problem at all of obtaining electrical meters and resistors of the requisite precision. The sound-power measurement consists of determining the mean steady-state acoustic energy density of the reverberant sound field by means of a calibrated pressure-sensitive microphone and of determining the rate of loss of acoustic energy from the room from the decay rate. The acoustic power of the source is then computed from the two data so obtained.

The frequency-range available to such sound-power measurements depends principally upon the size of the reverberation room. In the present case,

reasonably accurate measurements (about ±1.0 db or smaller) had been obtained in the course of previous research from about 100 cps to about 10,000 cps. The experiments with the model jets demonstrated a need to raise the upper frequency limit and it was found possible to extend the range to 20,000 cps when using instrumentation having a correspondingly wide frequency response.

Measurements at frequencies lower than 100 cps would require a larger reverberation room. The high-frequency limitation is caused mainly by the rapidly increasing absorption of the sound energy by the air in the reverberation room. A room with smaller dimensions might permit some extension to even higher frequencies while sacrificing the low-frequency end of the range. If a pair of rooms were used, one larger and one smaller than the existing room, the available frequency range might be extended by perhaps an octave at each end. There are, however, more factors to be considered than have been presented in this brief discussion of usable frequency range.

I.1 AIR FLOW FACILITIES

The air-flow system was a matter of considerable importance to this research program. Except for Reference 9, there was little precedence for introducing quantities of compressed air into a reverberation room. Also, the air had to be gotten out of the room again for several reasons, most important of which in this case, was to preserve microphone calibration which depends upon air density.

Consideration of the nature of the acoustical spectrum expected for simple jets and the selection of Mach one as a maximum flow velocity led to the one-half inch diameter nozzle and a mass flow rate of 100 SCFM. In order to concentrate the research effort upon the acoustical aspects, the air-flow system was kept as simple as possible. It was decided to monitor the mass flow rate of air upstream of the experimental nozzles and thus the air-flow system shown in Figure I.1 and Table I.I was selected.

The high-pressure air was supplied from outside the building by a truck-mounted compressor of the type used to operate pneumatic road-construction tools. Pressure regulation, temperature of the compressed air, and the cleanliness of the compressed air left something to be desired but these deficiencies merely constituted an inconvenience which did not appreciably degrade the experimental results. Certainly these deficiencies were not serious enought to justify the purchase of a high-quality stationary air supply for this one research program. For future research of the same type, the newer trailer-mounted rotary compressors look very promising and they are readily available in much larger capacities also. A somewhat higher performance filter and moisture trap would probably eliminate most of the remaining inconvenience. The remainder of the flow system functioned very well.

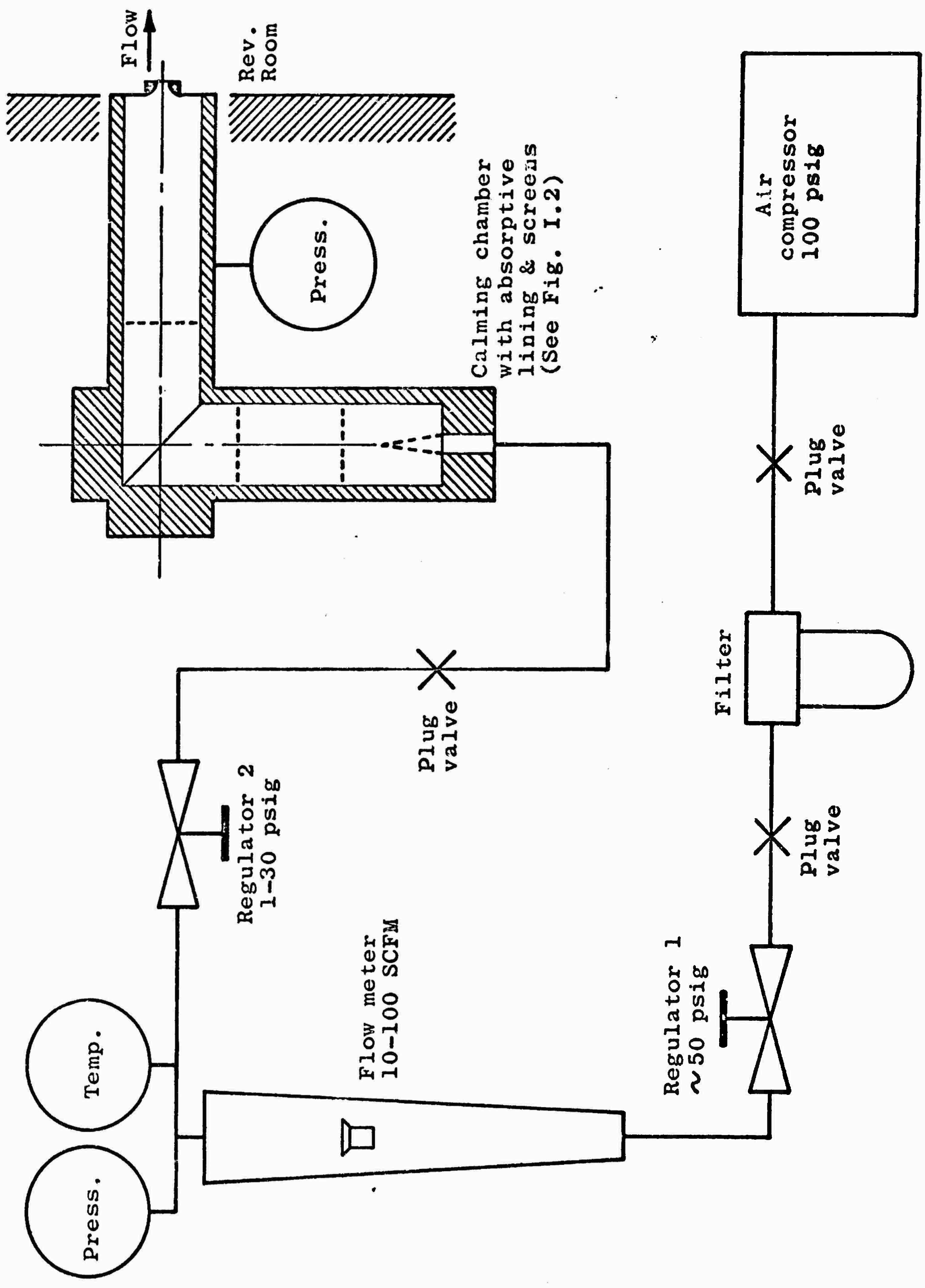


FIGURE I.1. AIR FLOW SYSTEM (See Table I.I)

TABLE I.I

COMPONENTS OF AIR FLOW SYSTEM

Air compressor:	Truck mounted reciprocating compressor, 105 SCFM, 100 psig.
Filter:	Schrader (A. Schrader's Son, Brooklyn, New York) No. 3336
Regulator 1:	Cash (A. W. Cash Co., Decatur, Illinois) Type 1000 HP-2, 3/4" regulating valve, 100 psig inlet, 50-80 psig spring adjustment range.
Flow meter:	Brooks (Brooks Instrument Co., Inc., Hatfield, Pennsylvania) Model 1110, Size 12, tube No. R-12M-25-4, float No. 12-RS-221, percent scale.
Pressure gauge:	Ashcroft (Manning, Maxwell & Moore, Inc., Stratford, Connecticut) Duragauge 4-1/2" -1279A, 0 - 100 psig.
Thermometer:	American (Manning, Maxwell & Moore, Inc.) Bimetal dial thermometer, Cat. No. 3-6360AH-S4, 30 - 130°F.
Regulator 2:	Cash Type 1000 LP-2, 3/4" regulating valve, 50-80 psig inlet, 1-30 psig spring adjustment range.
Pressure gauge:	Ashcroft Duragauge 4-1/2" -1279A, 0 - 30 psig.

Pressure regulator 1 was used to control the density of the compressed air at the flow meter. By properly setting the pressure there in accordance with the observed temperature (generally in the ranges of 60° to 130°F and 40 to 50 psig during these experiments) the flow meter could be made direct reading with a full-scale value of 100 SCFM. Industrial accuracy of ±2% of maximum scale was quite sufficient for present purposes and accuracies of ±0.5% are obtainable if needed. Pressure regulator 2, following the flow meter, reduced the pressure to the values needed at the model nozzles. A plug valve between regulator 2 and the calming chamber was partially closed for some conditions of small mass flow rate to provide regulator 2 with its minimum downstream pressure condition.

The calming chamber, shown in moderate detail in Figure I.2, constitutes an engineering design by acoustician's guess. Again, except for Reference 9, there is little in the literature to serve as a guide. It seemed essential to obtain quiescent air upstream of the experimental nozzle and to remove all flow noise or other noise which might arrive by way of the flow control system. The objective, of course, was to generate only simple jet noise when using smooth-approach nozzles. In other words, we needed to have solved the problems of jet silencing in order to know how to design the appropriate calming chamber. The chamber, illustrated in Figure I.2, was reasonably satisfactory for the intended purpose but it was large and heavy and, in addition, it provided for several contingencies which did not occur. Moreover, as has been mentioned in footnote 5, the sharp internal corner at the nozzle end may have produced some unintentional flow separation ahead of the nozzle under certain test conditions.

Now that the research has been accomplished, a much better calming chamber could probably be constructed with the internal acoustical silencing designed along the lines of Figure II.26. A new design of this type could lend itself to very thorough streamlining at the nozzle end of the chamber to minimize flow separation at or near the experimental nozzles.

The nozzle end of the calming chamber shown in Figure I.2 projected about 10 inches into the reverberation room through a slightly larger opening in the room wall. This opening around the chamber was calked with fiberglass and rags to close the opening while maintaining isolation against structure-borne noise. This arrangement of the chamber placed the jet noise source near the middle of an end wall of the reverberation room and fairly close to the wall. Separate experiments had demonstrated that the sound power was not affected detectably by locating the source this close to the wall. The axis of the chamber was angled slightly downward so that the exhaust flow would not impinge on the rotating vane.

The excess air escaped from the reverberation room through a one-foot square acoustically-lined duct about five feet long.[16] The pressure drop

[16]This duct was furnished free by the Acous-Trol Corporation, 19401 West McNichols Road, Detroit, Michigan.

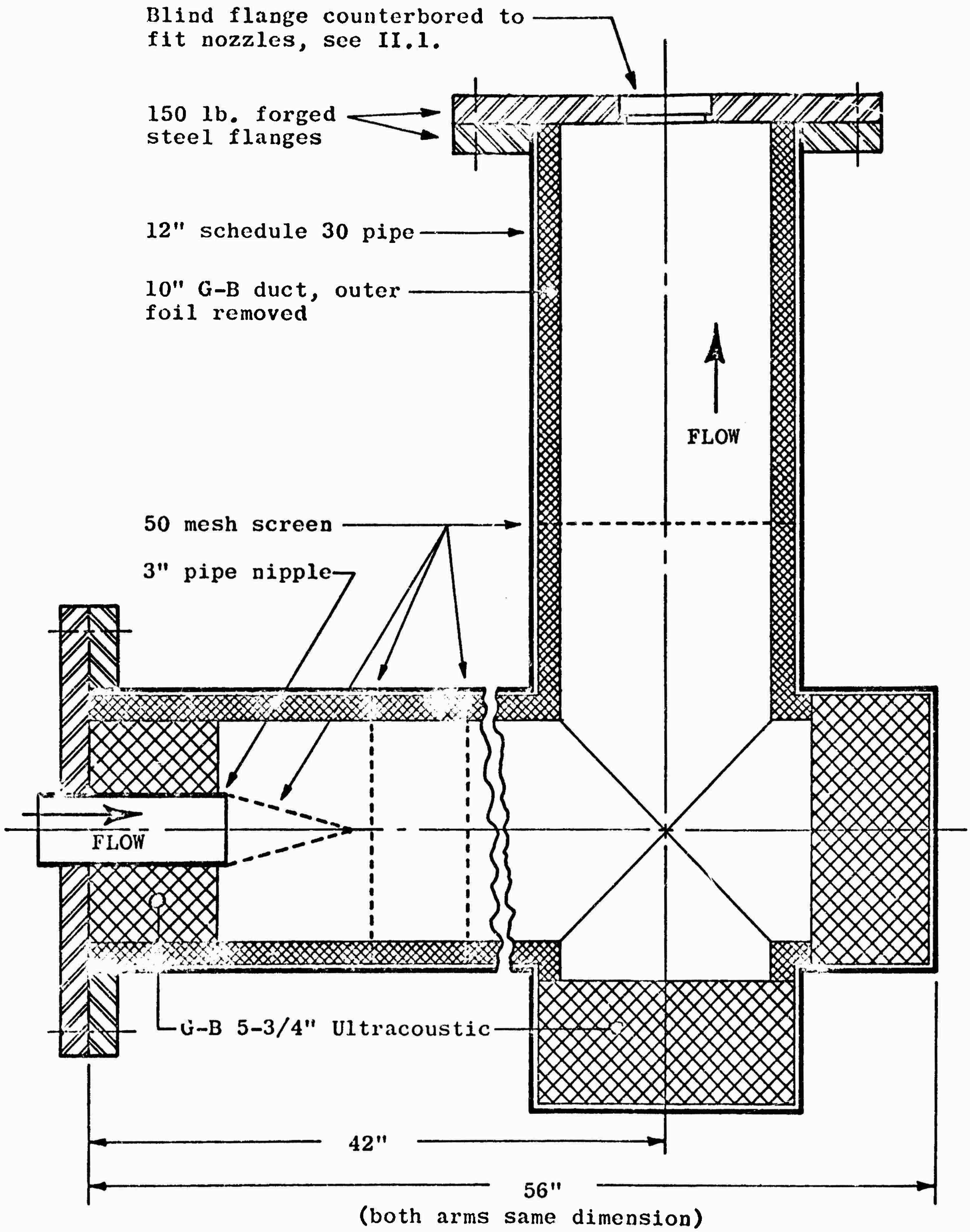


FIGURE I.2. CALMING CHAMBER (full sectional sketch)

across the duct was so small that essentially barometric pressure was maintained in the reverberation room. (This pressure drop was estimated to be about 0.005 inches of water.) The absorptive lining of the duct effectively uncoupled (acoustically) the reverberation room from the external space.

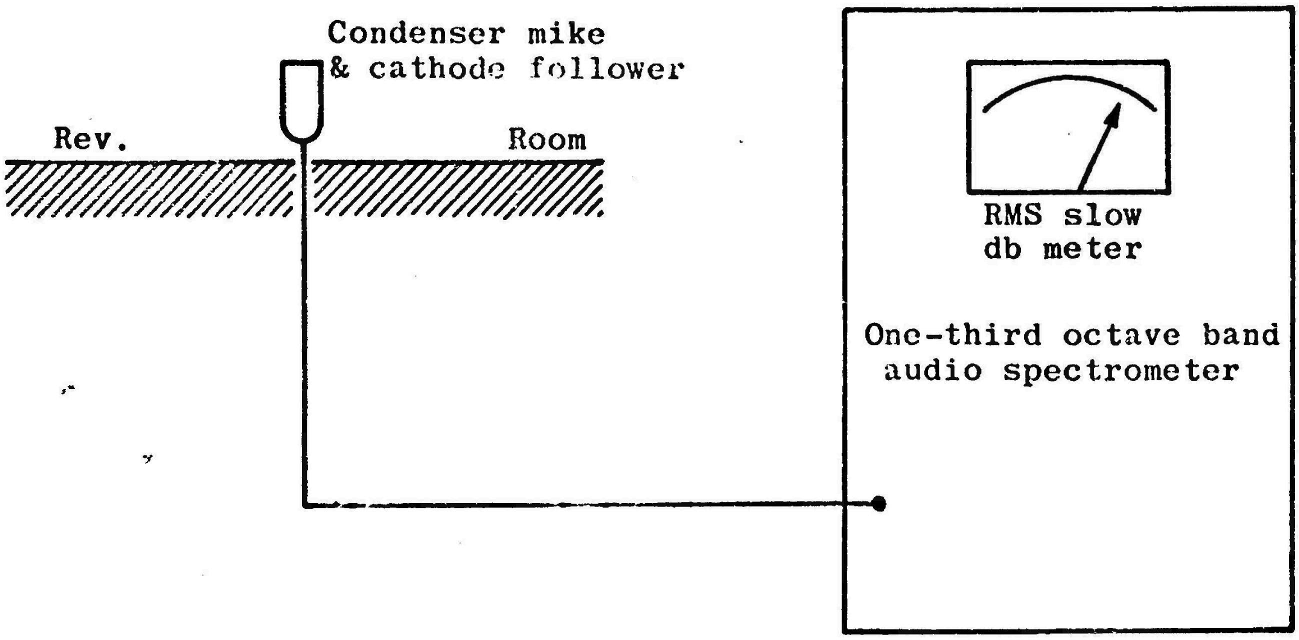
At the outset, we were concerned lest the air from the jet interfere in some way with the prerequisite atmospheric conditions within the reverberation room. As it turned out, no difficulties of this type were encountered except for some moderate changes in humidity and these effects were easily taken into account by frequent redetermination of the decay rates at the higher frequencies. Estimating, it would appear feasible to work with cold air flow rates up to perhaps 400 or 500 SCFM in this same reverberation room. The use of heated air or air containing an appreciable percentage of combustion products has not been tried. Extension of the research in these directions should be tried in separate experiments but a cautious approach is urged to preserve the high accuracy and reproducibility of the reverberation-room methods.

I.2. ACOUSTICAL MEASUREMENT SYSTEM

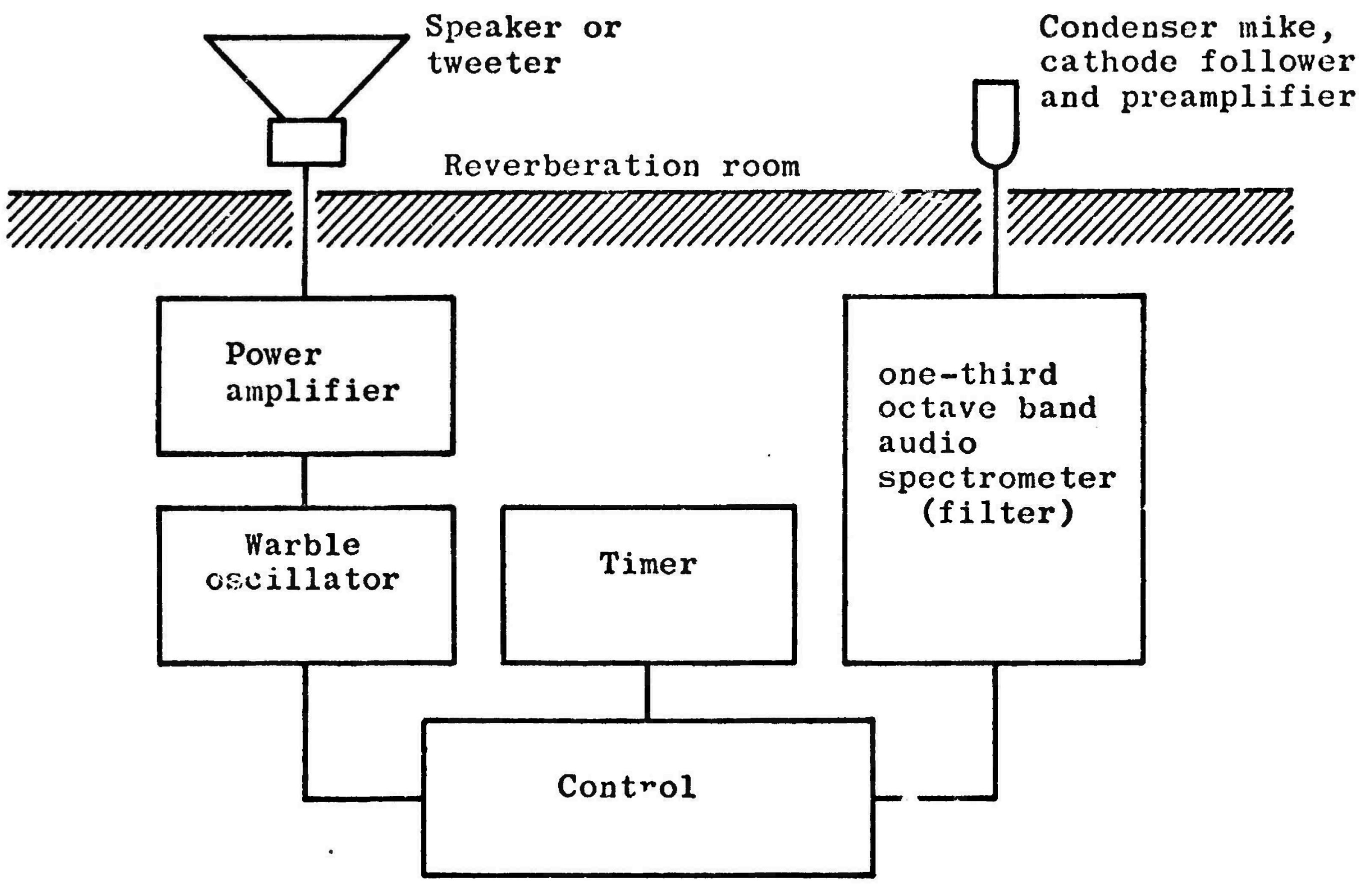
The acoustical measuring system was conventional for reverberation room applications and is described in Figure I.3 and Table I.II. A one-half inch diameter condenser microphone possessing a flat diffuse-field frequency response characteristic was selected for convenience in adjusting the data to sound pressure level and its compatibility with the spectrometer. When it became necessary to extend the high-frequency measurements to 20,000 cps, some variation in microphone sensitivity from band to band became involved but still only of small magnitude. The sound pressure measurements were taken with the microphone mounted stationary in a part of the reverberation room remote from the model jet and with the microphone spaced well away from the walls and rotating vane.

The manufacturer's values for microphone sensitivity were checked by the diffuse-field reciprocity method, an absolute method, and agreement to within about ±0.2 db was obtained. This result was more than adequate for the purposes of this research and so the calibration values furnished by the manufacturer were used for all data reduction. Actually, as a result of an accident to one condenser microphone, a second microphone of the same type was used for about half of the measurements. Check sound power measurements taken with the bare one-half inch diameter smooth-approach nozzle, Figure II.1, repeated within a fraction of a decibel. Many such direct and indirect results led to considerable confidence in both the relative and absolute accuracy of the acoustical measurements.

Electrical checks of the spectrometer system demonstrated it to be well within manufacturer's specifications. As indicated, the output of the spectrometer was read from its meter operating in the slow (speed of response) rms



A. Sound Pressure Instrumentation



B. Decay Rate Instrumentation

FIGURE I.3. ACOUSTIC INSTRUMENTATION (See Table I.II)

TABLE I.II

COMPONENTS OF ACOUSTIC INSTRUMENTATION

A. Sound Pressure Instrumentation

Microphone:	Bruel & Kjaer (B & K Instruments, Inc., Cleveland, Ohio) Type 4134 1/2" diam condenser microphone Type 2615 cathode follower
Spectrometer:	Bruel & Kjaer Type 2142 Audio Frequency Spectrometer

B. Decay Rate Measurements

Warble oscillator:	Bruel & Kjaer Type 1014 Beat Frequency Oscillator
Power amplifier:	McIntosh (McIntosh Laboratory Inc., Binghamton, New York) Type MC 30
Loudspeakers:	GE (General Electric Co., Auburn, New York) Model 1201 B 12" wide range speaker (used below 4000 cps) Electro-Voice (Electro-Voice, Inc., Buchanan, Michigan) Type T-350 VHF Tweeter (used above 4000 cps)
Microphone:	Bruel & Kjaer Type 4134 and 2615 (battery operated)
Preamplifier:	Tektronix (Tektronix, Inc., Portland, Oregon) Type 122 Low-level preamplifier
Filter:	Bruel & Kjaer Type 2142 Audio Frequency Spectrometer
Timer:	Beckman (Beckman Instruments, Inc., Richmond, California) Model 7360R Universal Eput & Timer
Control:	Special laboratory-built circuitry

mode. It was, however, necessary to average the meter fluctuations by eye for a period of ten to thirty seconds. The meter fluctuations were wildest at low frequencies and since the divisional spacing of the meter's decibel scale was not uniform, the visual averaging process placed considerable stress upon the experimenter. In future experiments requiring a similar collection of large amounts of data, it may be worthwhile to use electronic integration, digital readout, and perhaps reduction of data by digital computer for speed and convenience.

In the case of the decay-rate measurements, both the microphone and the loadspeaker were mounted on the rotating reflector vane and moved with it. Past experience has shown this arrangement to be most satisfactory, at least in our reverberation room. Originally, a dynamic microphone was used for decay-rate measurements. Its frequency response did not extend to 20,000 cps and so a one-half inch diameter condenser microphone was used ultimately for the decay-rate measurements also. Battery operation of this condenser microphone became necessary to avoid elaborate rewiring of the electronic instrumentation mounted on the rotating vane.

A warble-tone signal source was used because it was the most convenient with the instrumentation at hand. Filtered one-third octave bands of noise would have been equally satisfactory if a second spectrometer had been available or if a special send-receive high-speed switching system had been developed to time-share the one spectrometer.

The time for the acoustic signal in the reverberation room to decay between preset levels was measured with an electronic time-interval meter over most of the frequency range. Laboratory-built control circuitry permitted automatic accumulation of the time for twenty decays in each frequency band. These twenty decays represented effectively a spatial-average value for the reverberation room as a consequence of the vane rotation. A second set of twenty decays was obtained immediately following the first, and if the values did not agree within 1%, additional data were collected to provide a more satisfactory average value.

A high-speed level recorder was utilized for decay-rate measurements above 8,000 cps. This method became necessary because the electronic time constants of the control circuitry were not quite short enough to accommodate the very fast decay rates encountered at the highest frequencies and we did not want to take the time to revise the control circuitry for faster operation. At lower frequencies, the electronic time-interval meter method and the high-speed level recorder method gave consistent results but data collection was much more convenient using the time-interval meter.

In the future, it would be convenient to provide a multi-channel decay-rate measuring system so that measurements can be made practically simultaneously in several frequency bands. The present single-channel operation is rather time consuming. Particularly, if more rapid fluctuations in

humidity are permitted because of using larger mass flow rates of compressed air, faster collection of decay-rate data may become essential at the higher frequencies. Multi-channel instrumentation would appear to present no essential difficulties beyond the cost of the component instruments.

O.P.

APPENDIX II

NOZZLES AND TEST CONFIGURATIONS

This appendix is intended to clarify, by means of sketches, the geometry of the various nozzles and configurations of objects tested.

The several nozzles were designed to fit flush with the interior face of the blind flange on the nozzle end of the calming chamber. (See Figure I.2) This end of the chamber projected about 10 inches into the reverberation room through an opening in the wall and was vibration isolated from it. The nozzles were each provided with a shoulder which insured the proper positioning in the flange and an O-ring produced an air-tight seal. (See Figure II.1) The nozzles and test configurations were held in place by external bolts, clamps, and dogs which were located far outside of the air flow paths; these accessories have been omitted from the several sketches.

In general, the nozzles and object configurations had circular symmetry and the corresponding sketches are in full section except where noted. The air flow path is always from left to right.

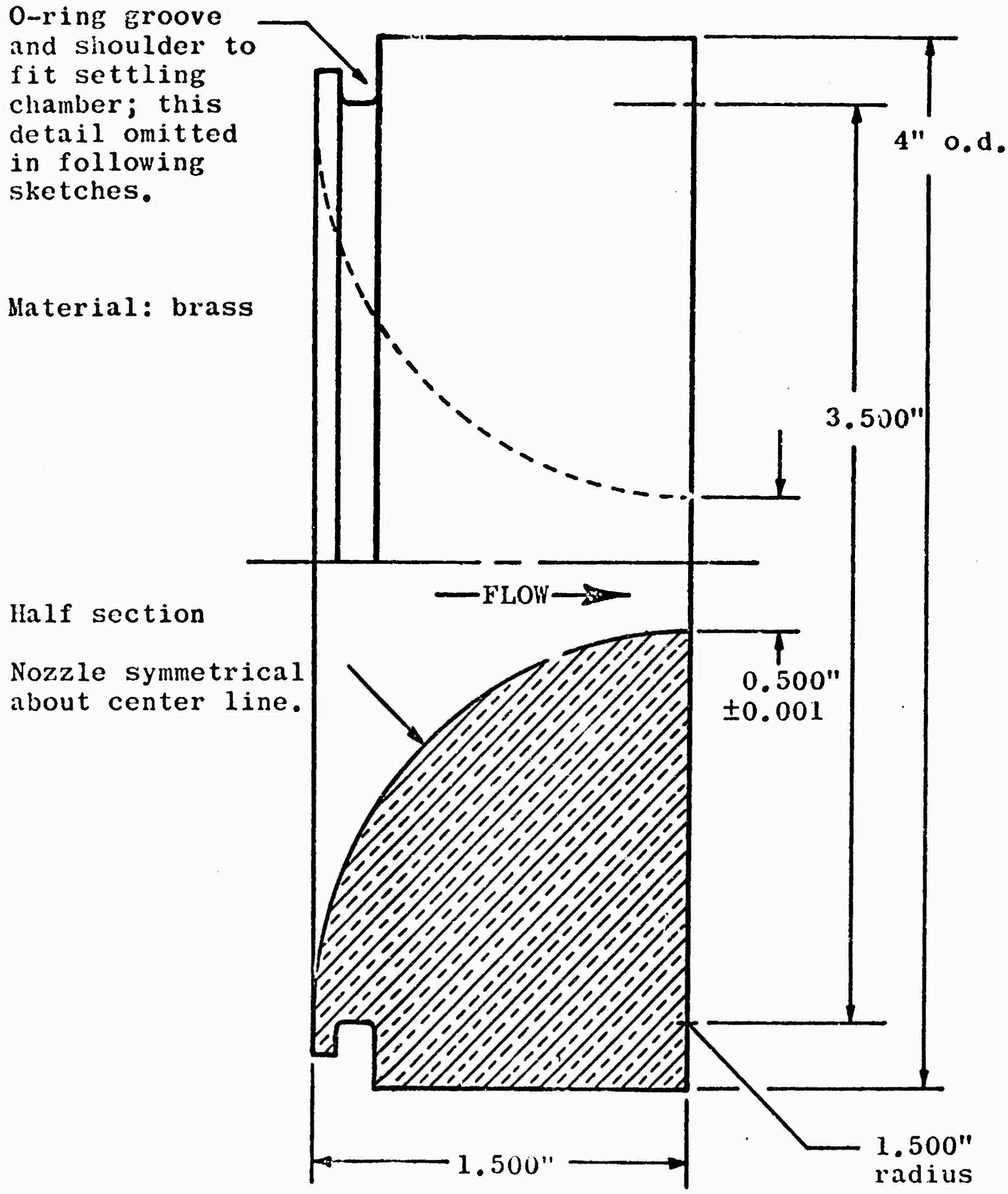


FIGURE II.1. 0.500 INCH DIAMETER SMOOTH APPROACH NOZZLE.

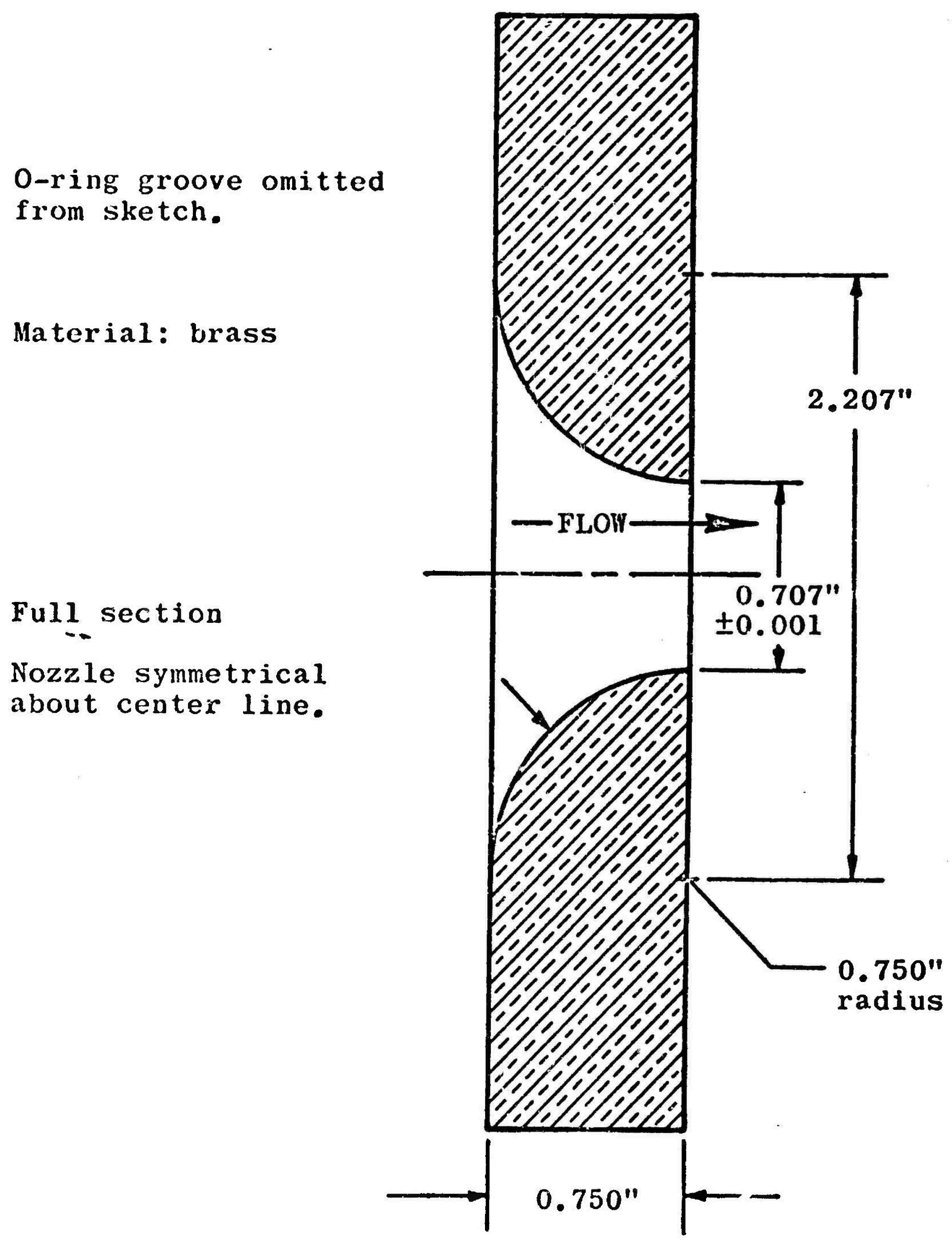


FIGURE II.2. 0.707 INCH DIAMETER SMOOTH APPROACH NOZZLE.

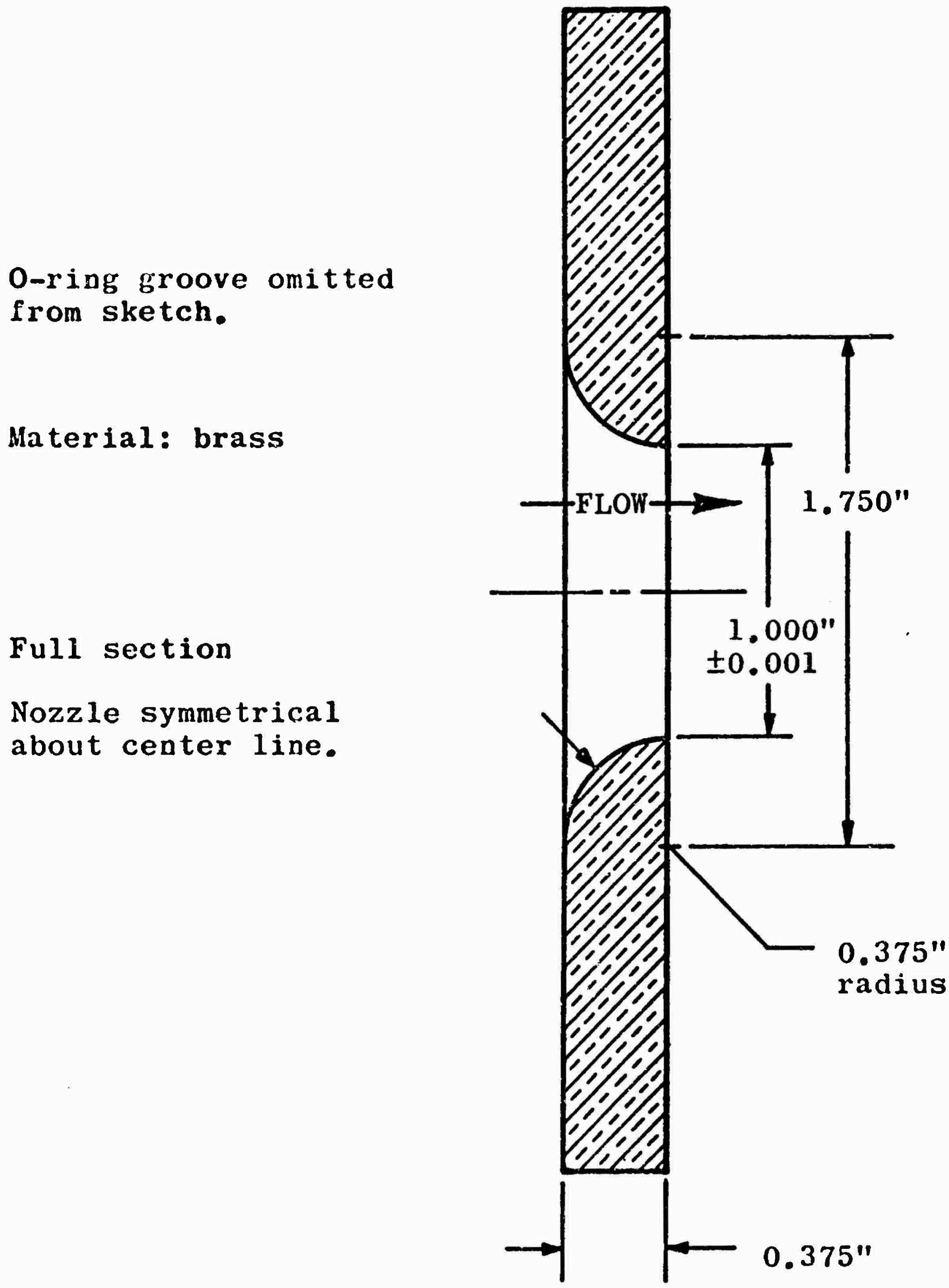


FIGURE II.3. 1.000 INCH DIAMETER SMOOTH APPROACH NOZZLE.

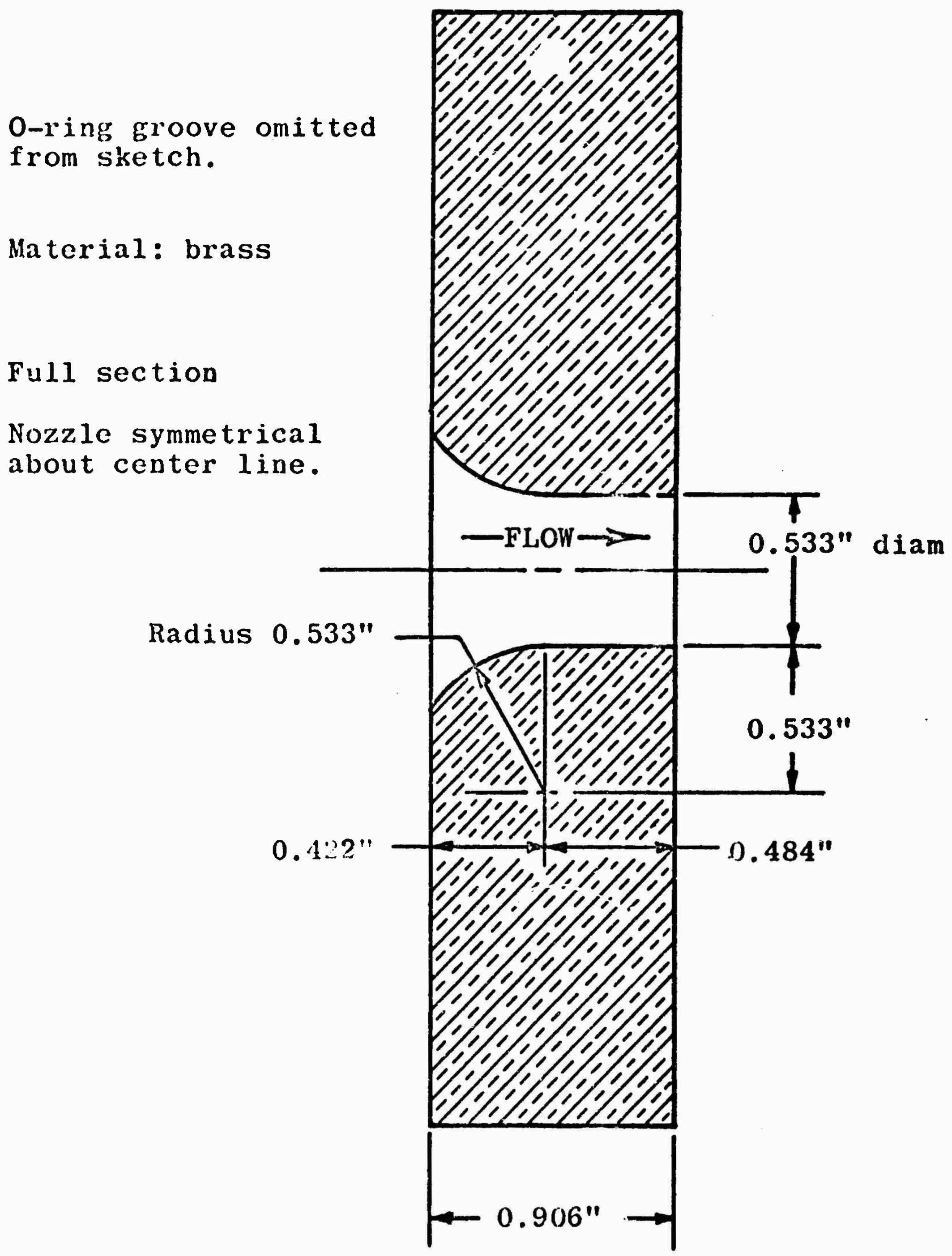


FIGURE II.4. DUPLICATE OF SPERRY'S NOZZLE NO. 100 (Ref. 3).

O-ring groove omitted from sketch.

Material: brass

Full section

Nozzle symmetrical about center line.

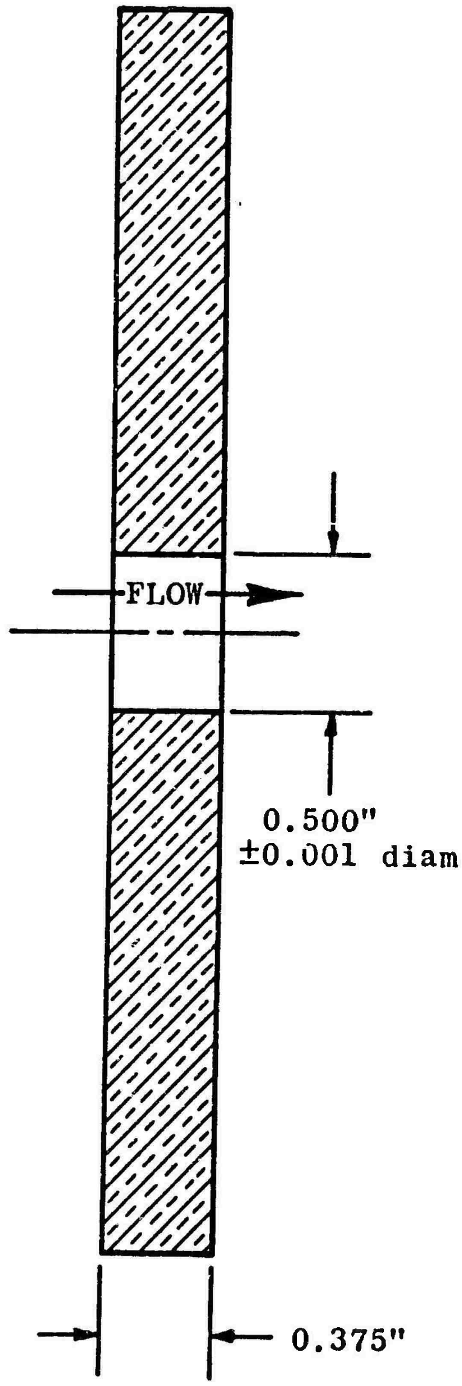


FIGURE II.5. 0.500 INCH DIAMETER SHARP EDGE NOZZLE.

O-ring groove omitted from sketch.

Material: brass

Full section

Nozzle symmetrical about center line.

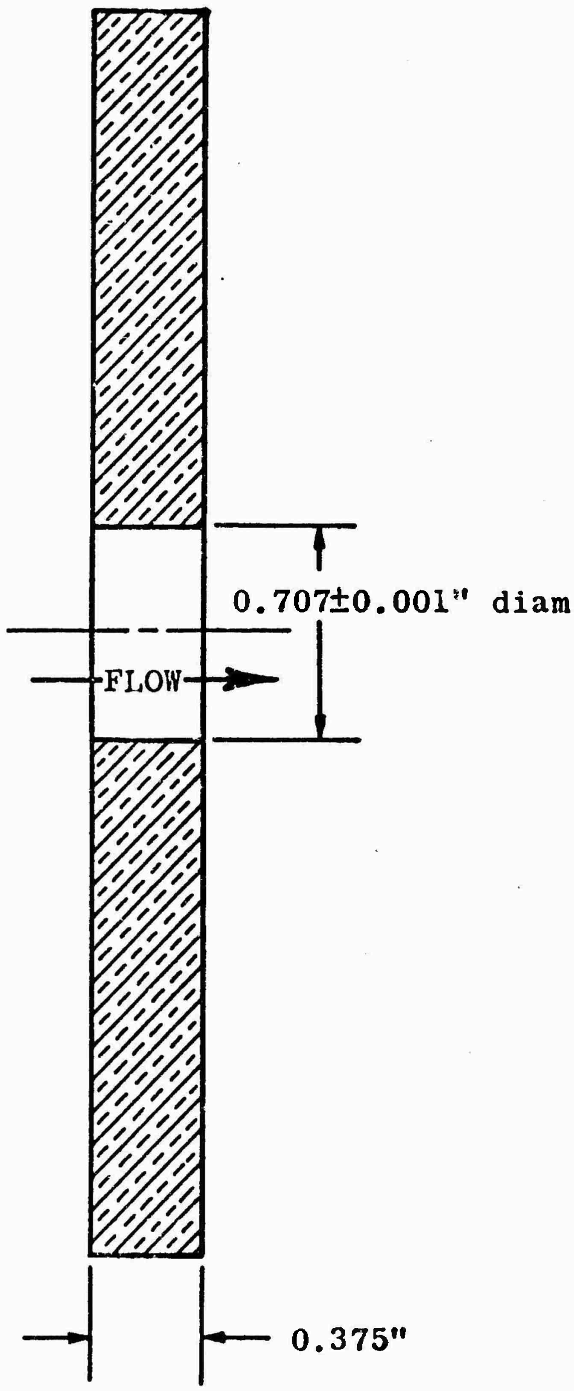


FIGURE II.6. 0.707 INCH DIAMETER SHARP EDGE NOZZLE.

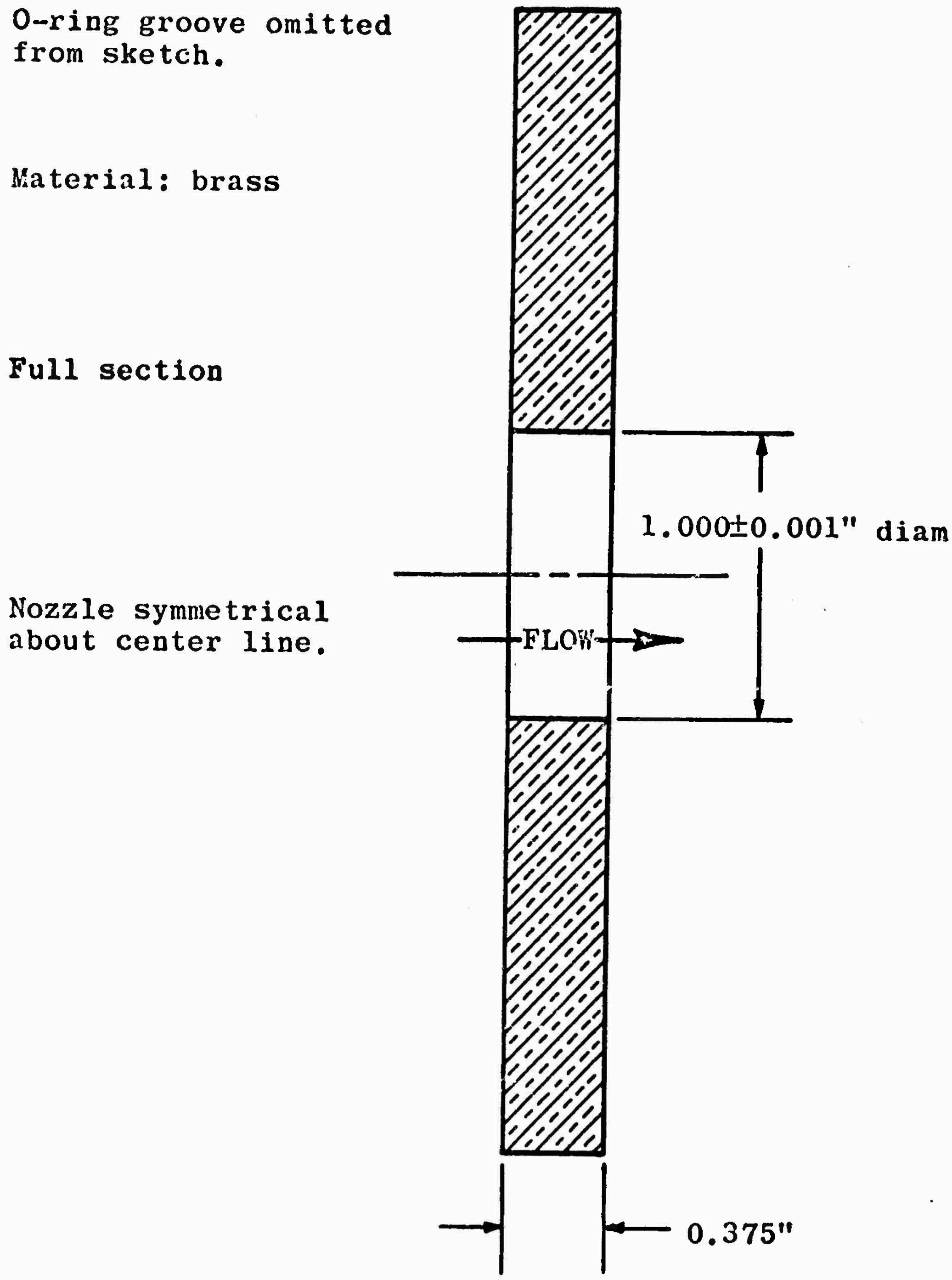


FIGURE II.7. 1.000 INCH DIAMETER SHARP EDGE NOZZLE.

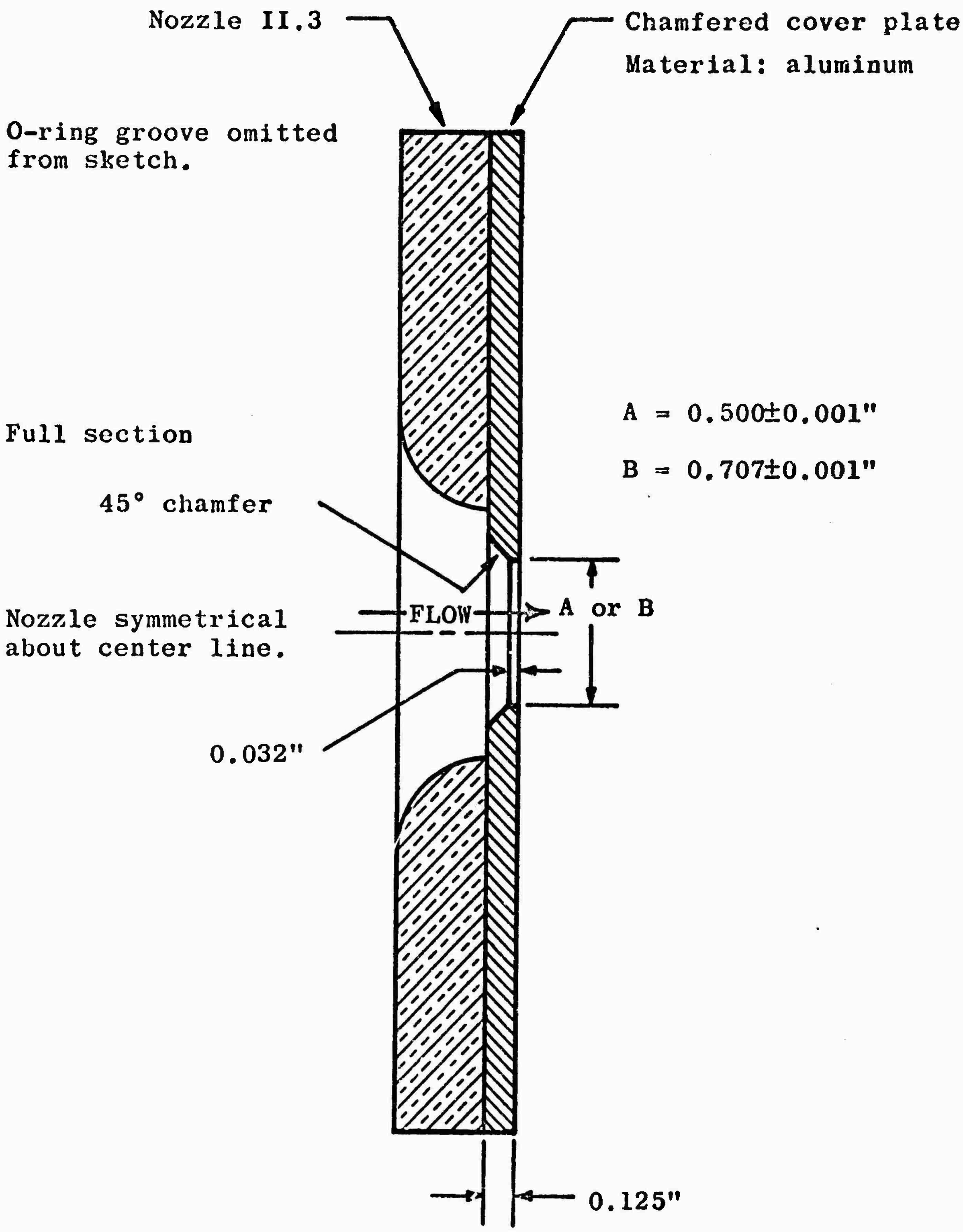
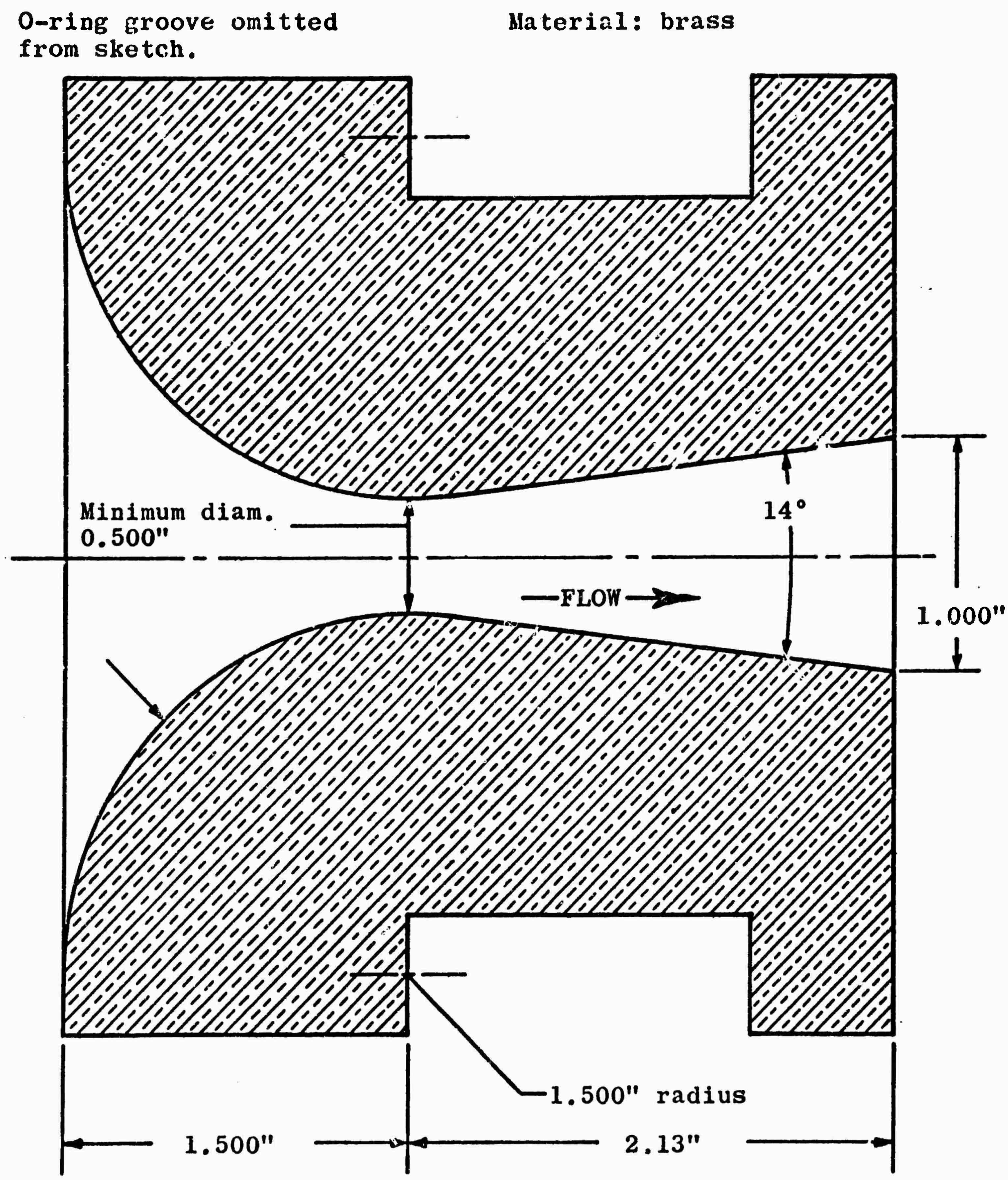


FIGURE II.8. NOZZLE II.3 WITH CHAMFERED COVER PLATE.



Note: Approach portion of this nozzle is identical in design to nozzle II.1. Nozzle symmetrical about center line.

FIGURE II.9. DIFFUSER FROM 0.500 TO 1.000 IN DIAMETER.

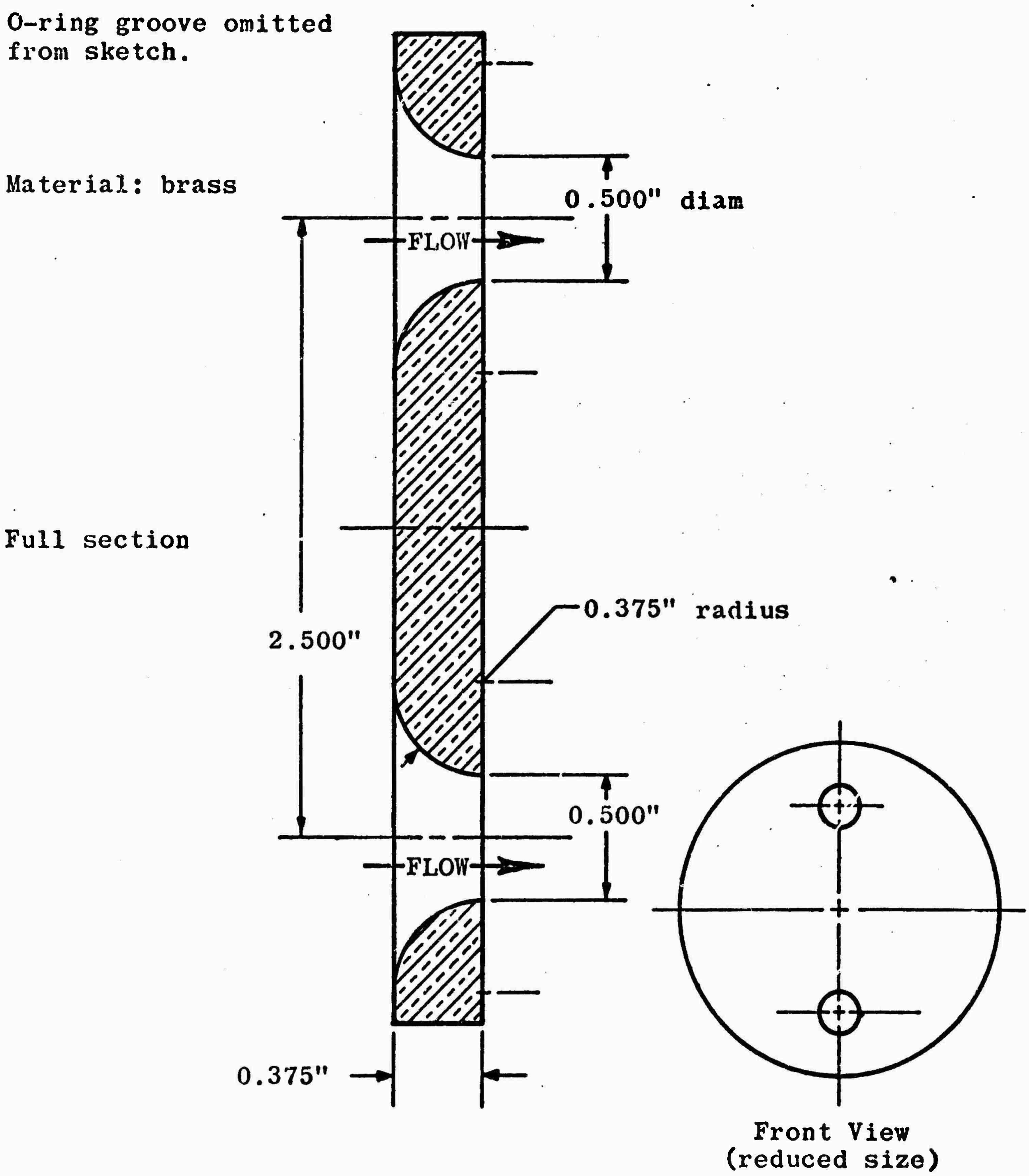


FIGURE II.10. TWO 0.500 INCH DIAMETER NOZZLES 2.500 INCHES ON CENTERS.

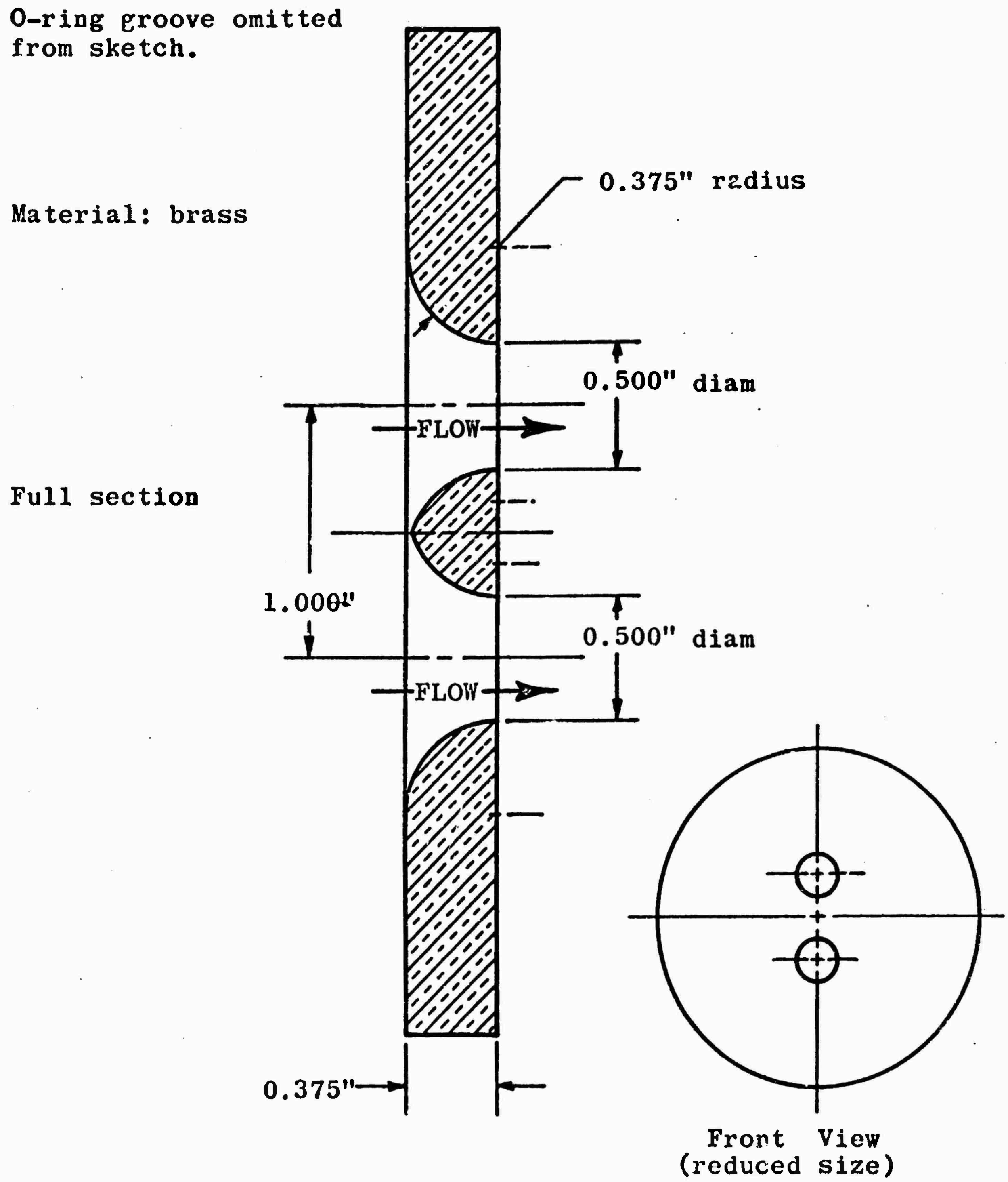


FIGURE II.11. TWO 0.500 INCH DIAMETER NOZZLES 1.000 INCH ON CENTERS.

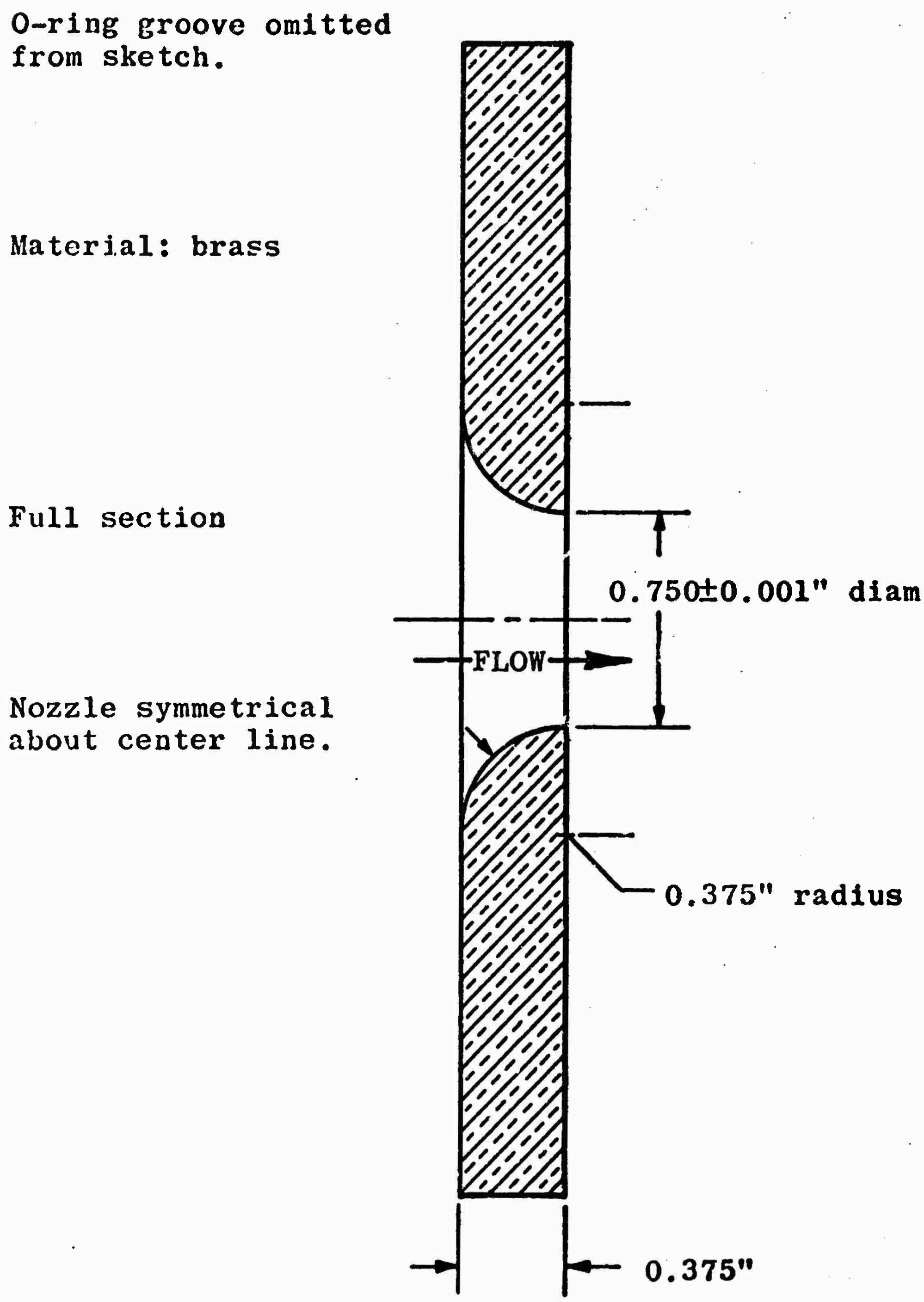


FIGURE II.12. 0.750 INCH DIAMETER SMOOTH APPROACH NOZZLE.

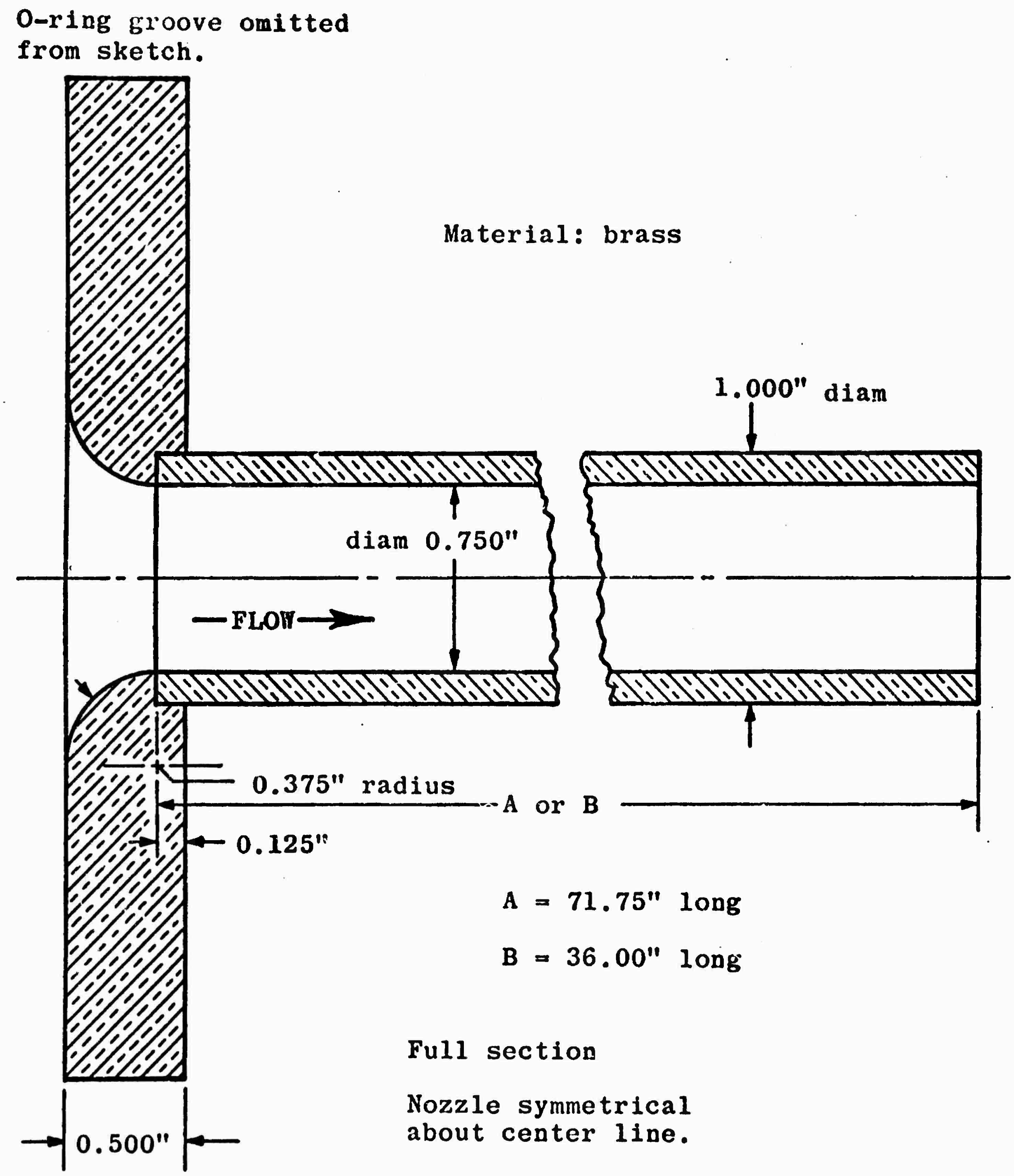


FIGURE II.13. NOZZLE WITH LONG EXTENSION TUBE.

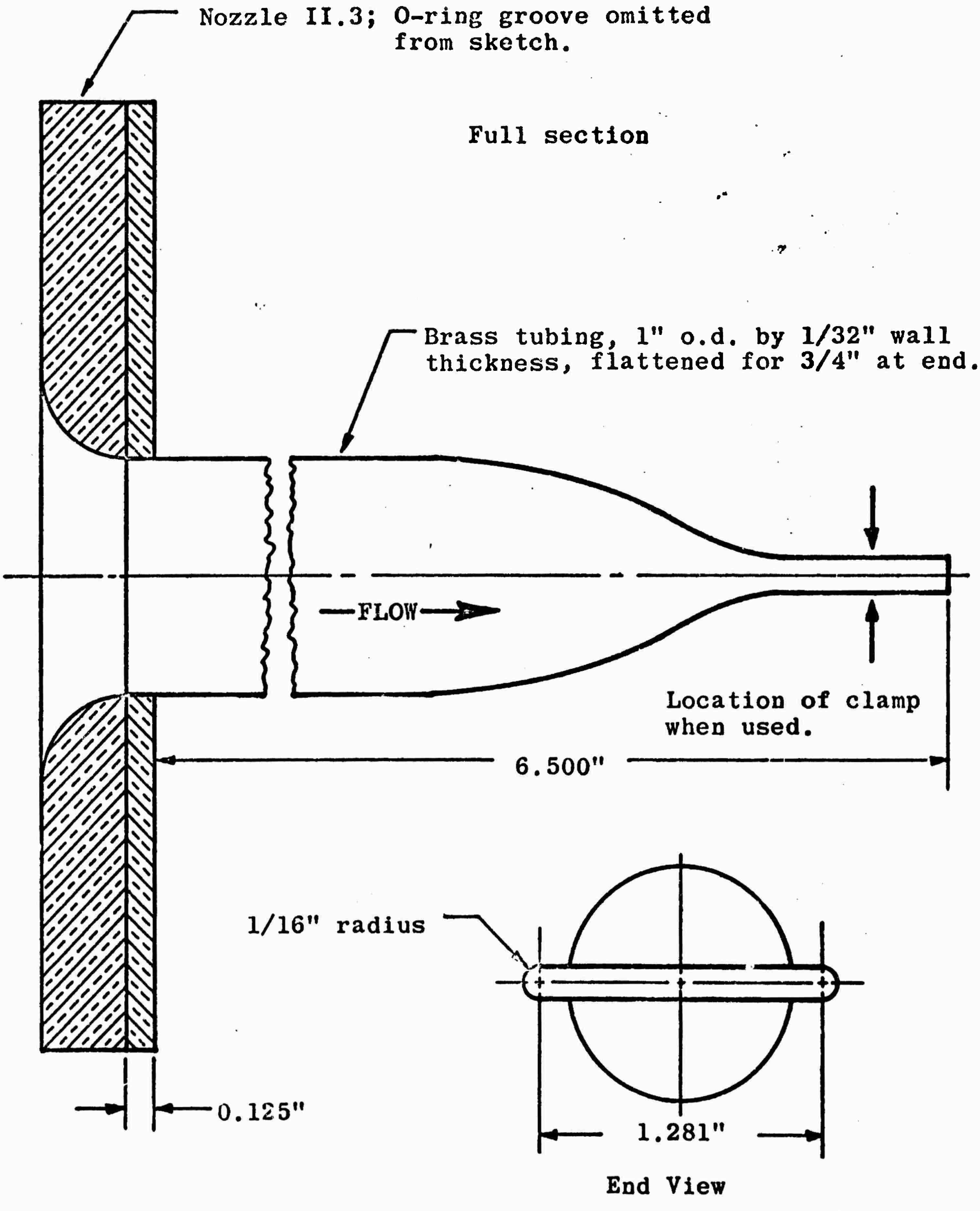


FIGURE II.14. EXTENSION TUBE WITH FLATTENED END.

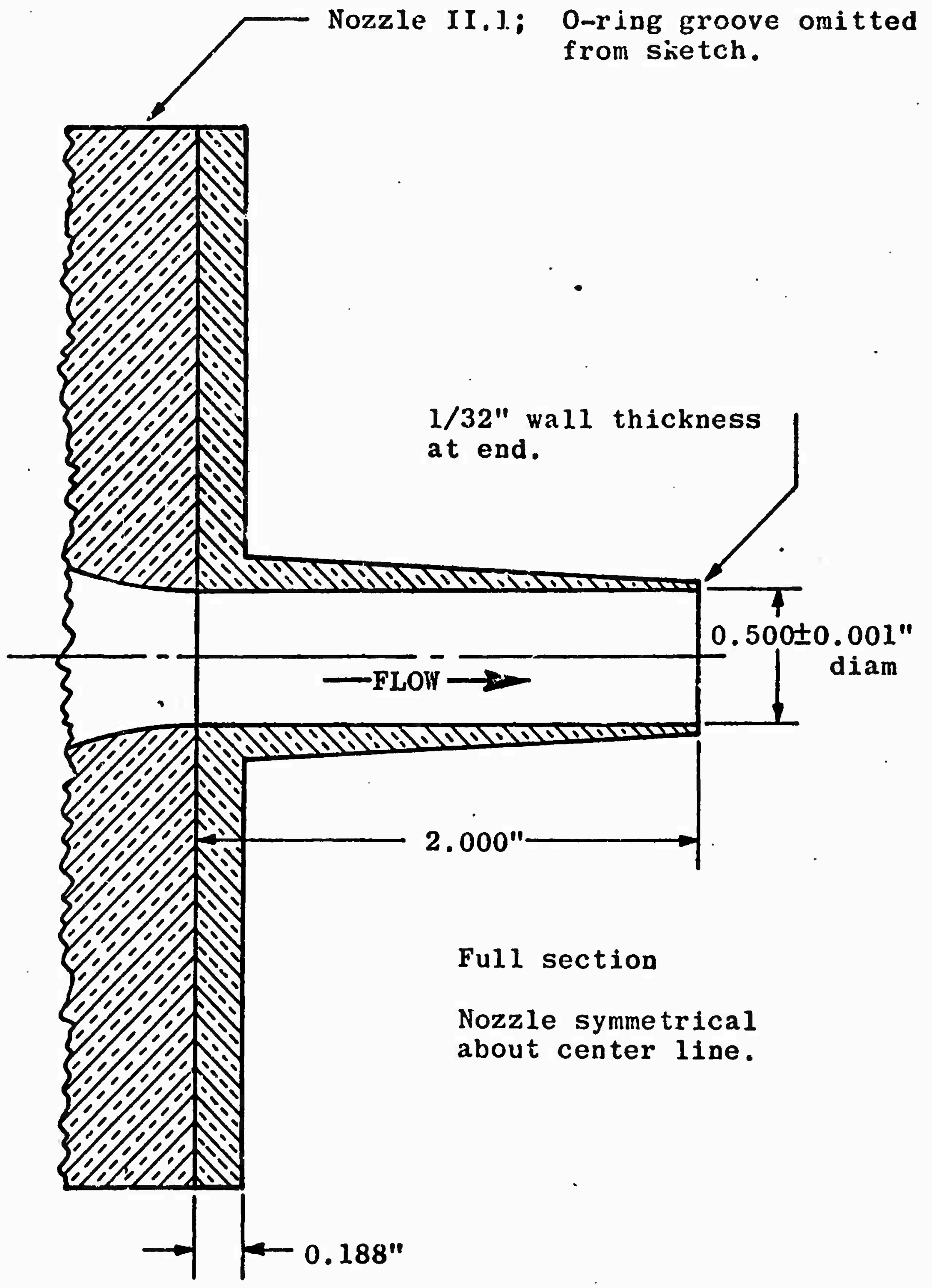
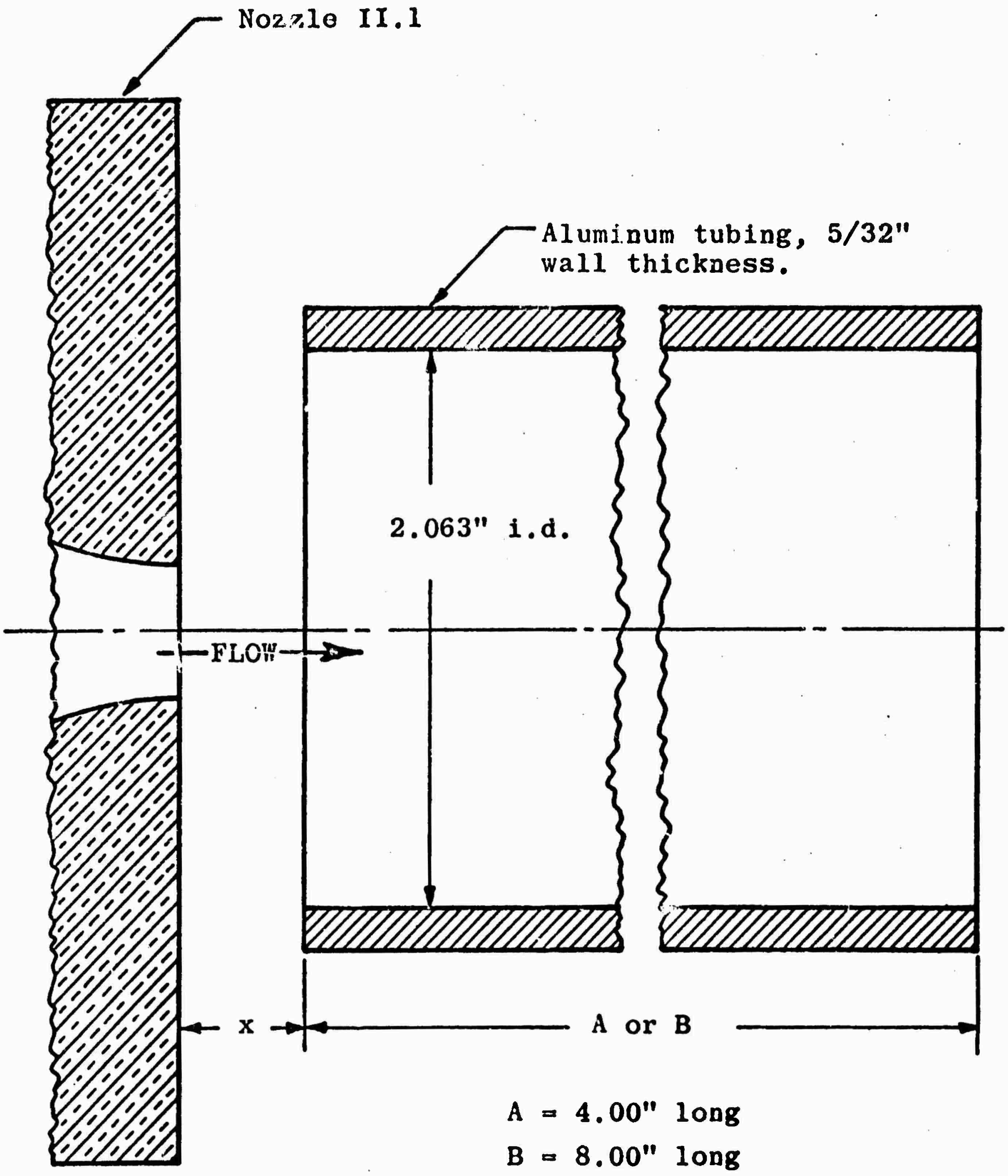


FIGURE II.15. SHORT EXTENSION TUBE.



Full section. Nozzle symmetrical about center line.

FIGURE II.16. STRAIGHT TUBE SURROUNDING JET.

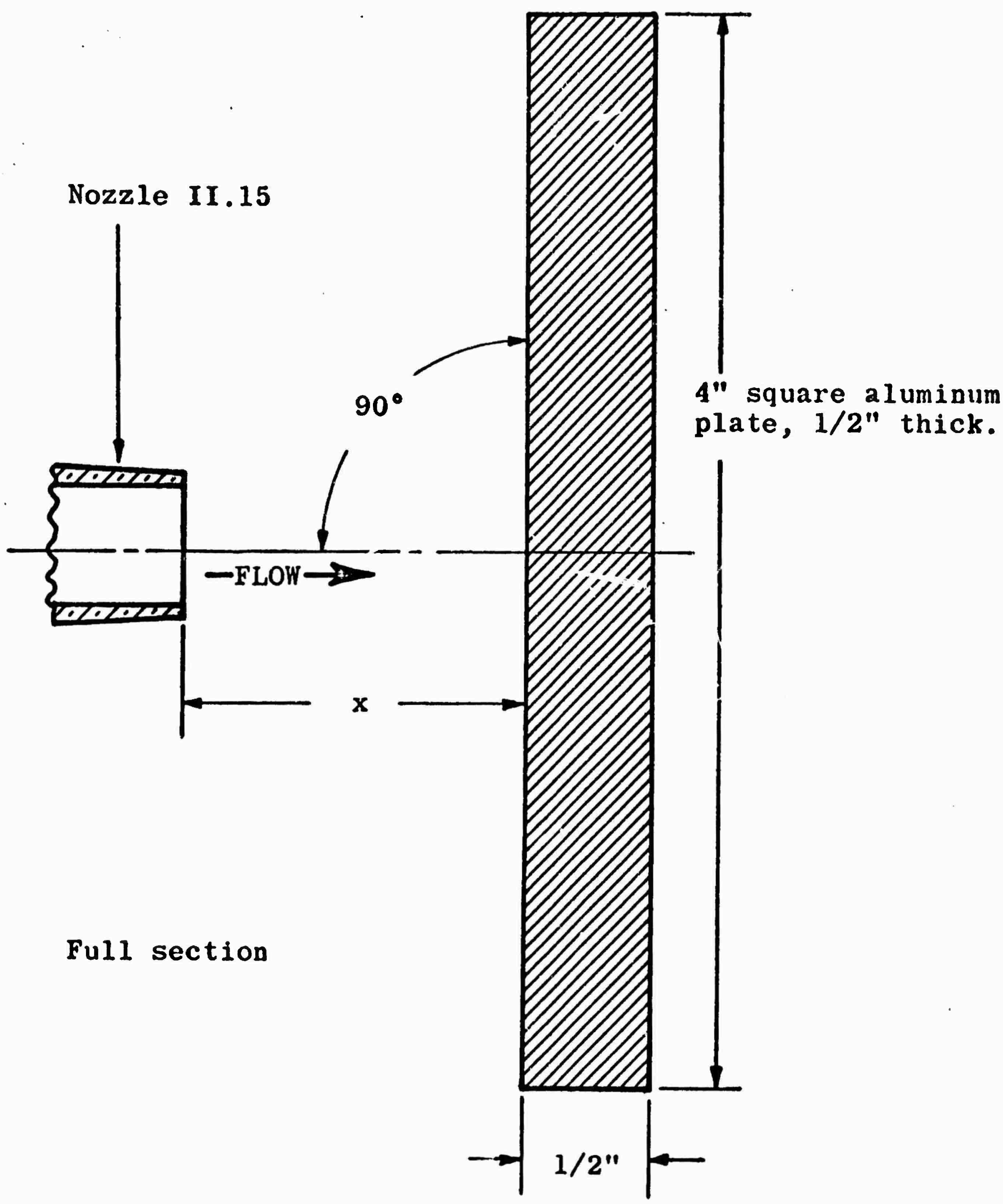


FIGURE II.17. FLAT PLATE PERPENDICULAR TO JET.

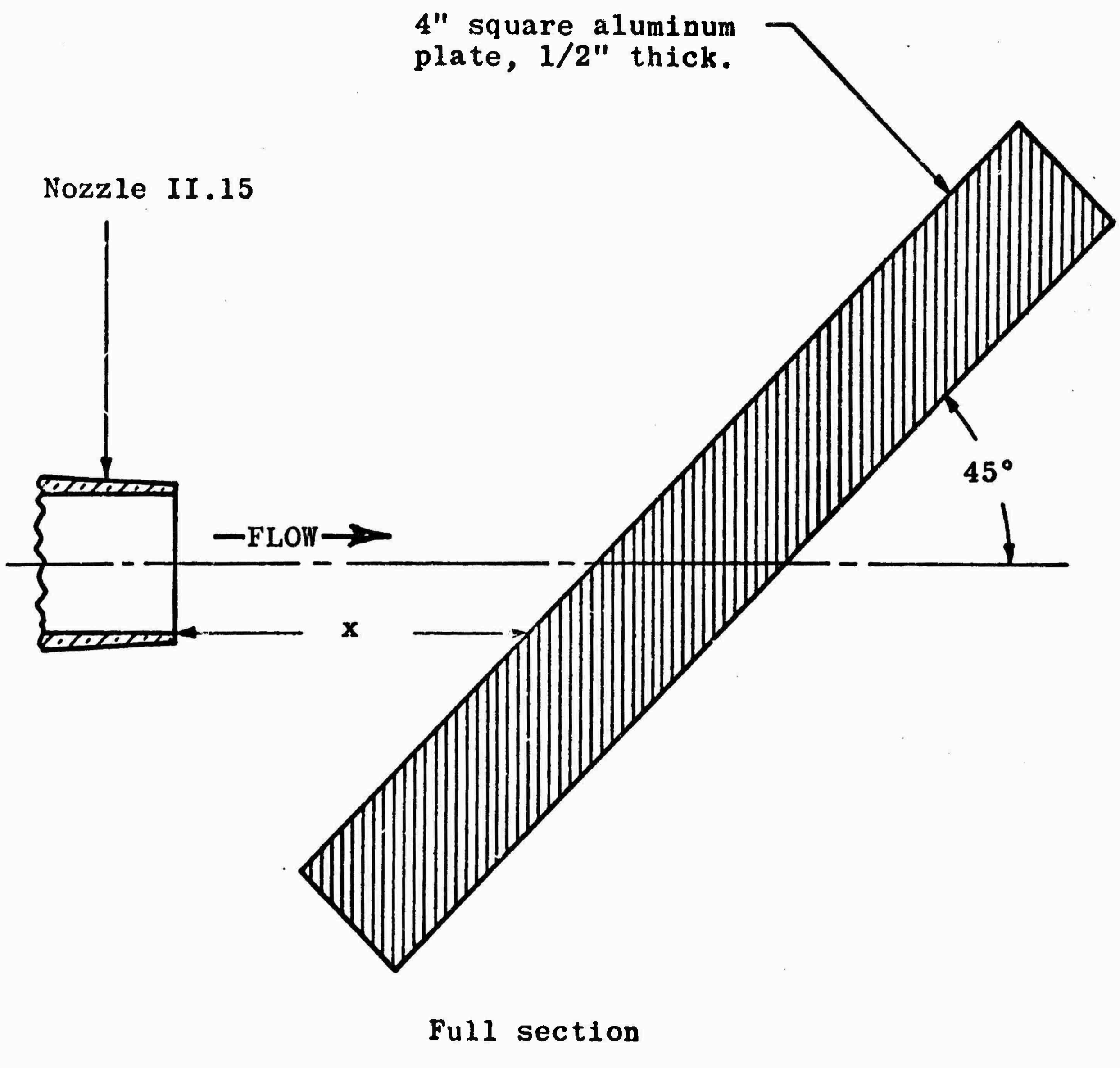
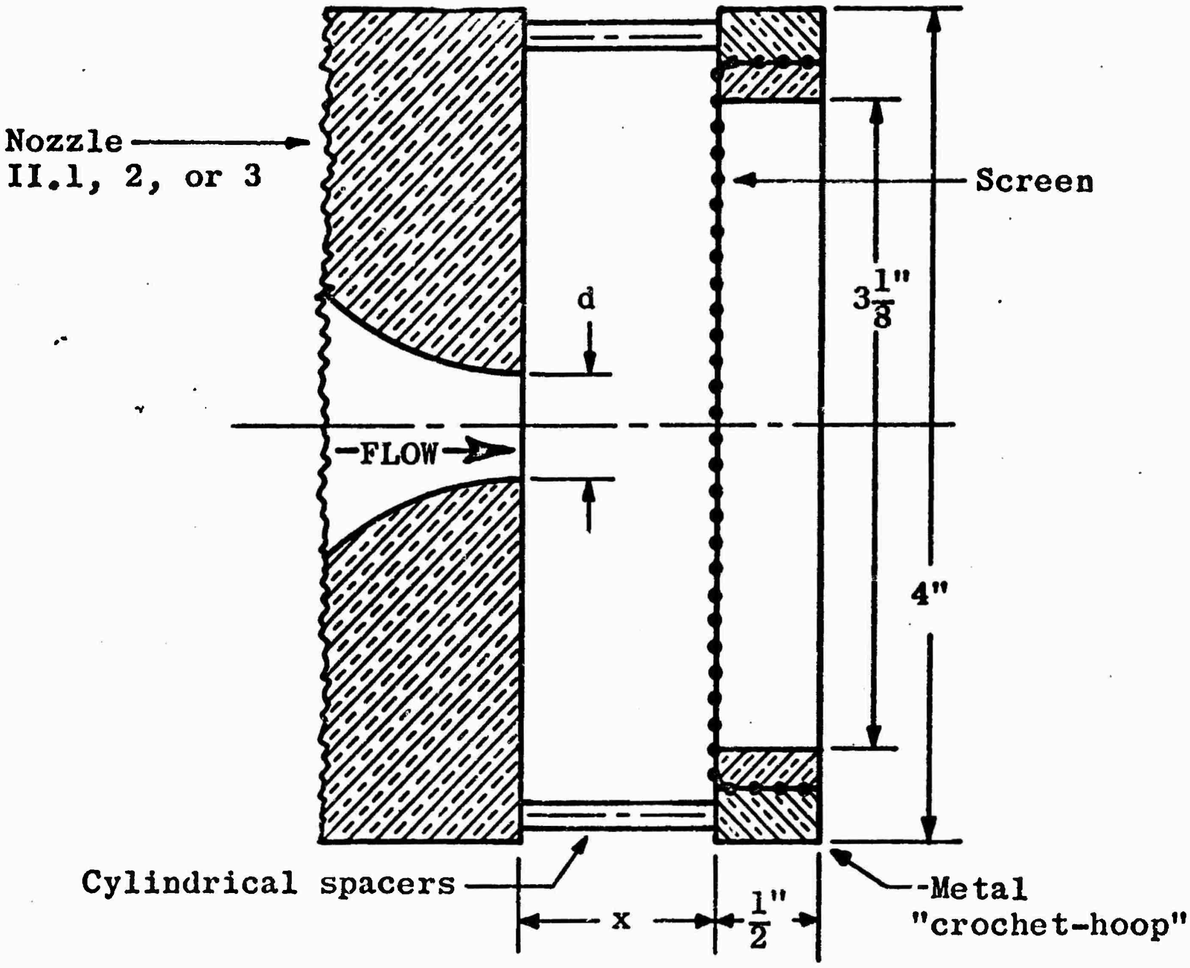
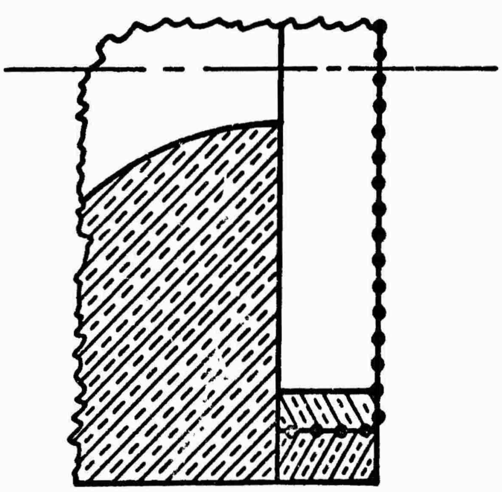


FIGURE II.18. FLAT PLATE AT ANGLE TO JET.

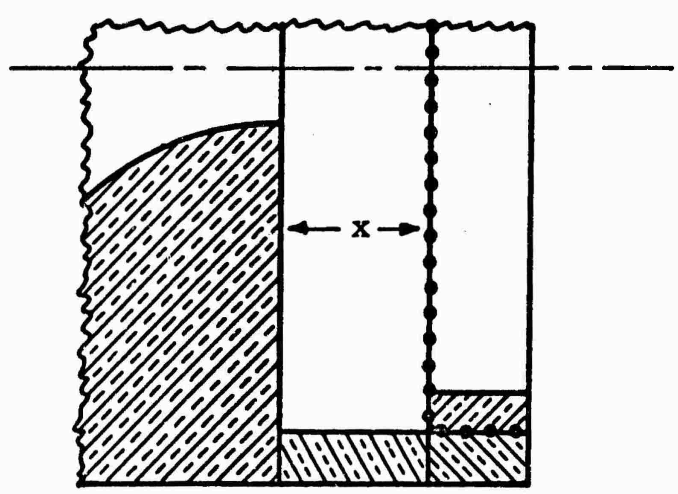


II.19A - Screen supported on cylindrical spacers (Side flow permitted)

II.19B - Screen soldered to 6" square frame and clamped in position. (Side flow permitted)



II.19C - Screen spaced by own ring. (Side flow not permitted)



II.19D - Screen spaced by complete spacer ring of required thickness x. (Side flow not permitted)

FIGURE II.19. SCREENS PERPENDICULAR TO JET AXIS.

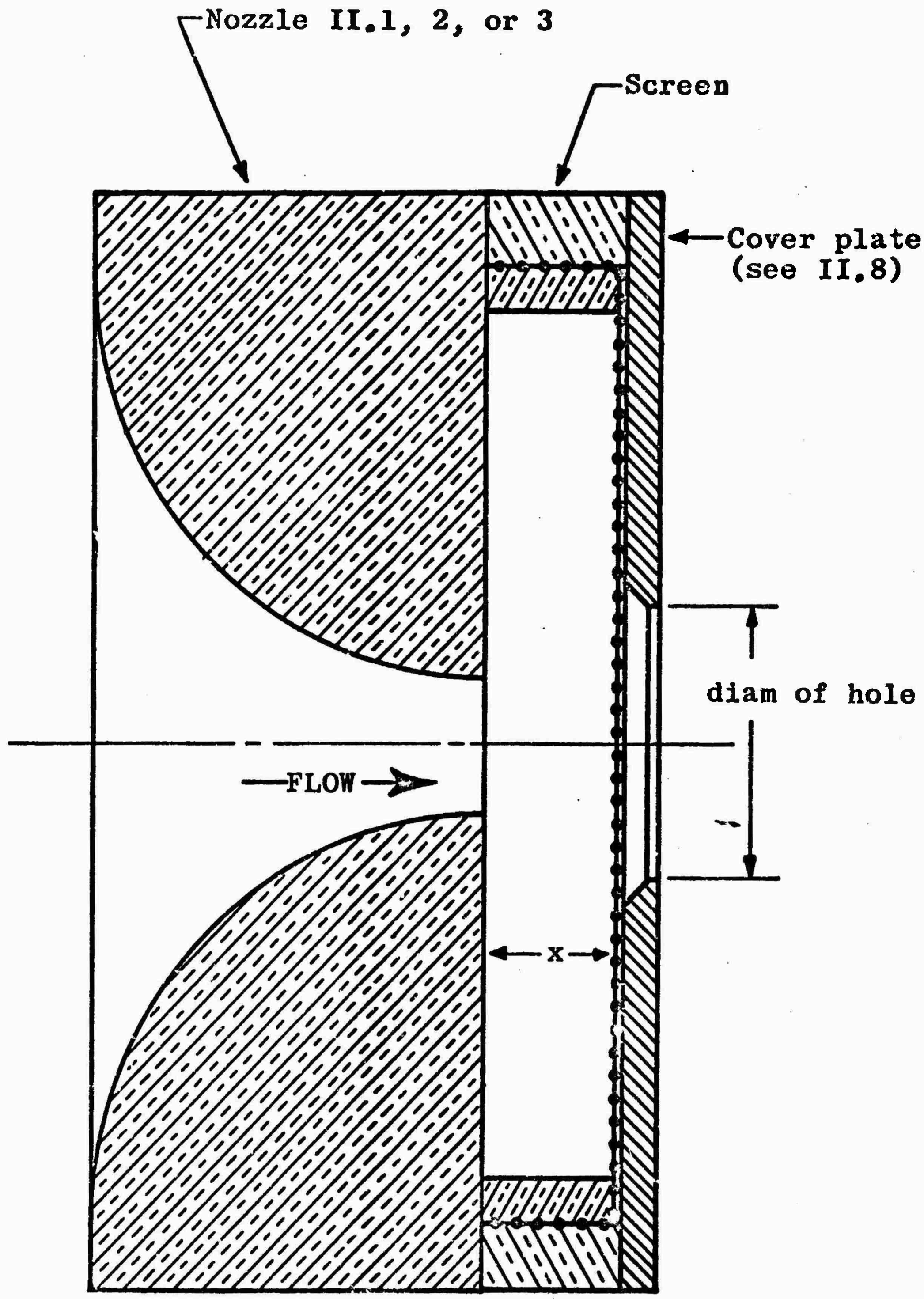


FIGURE II.20. SCREEN WITH COVER PLATE

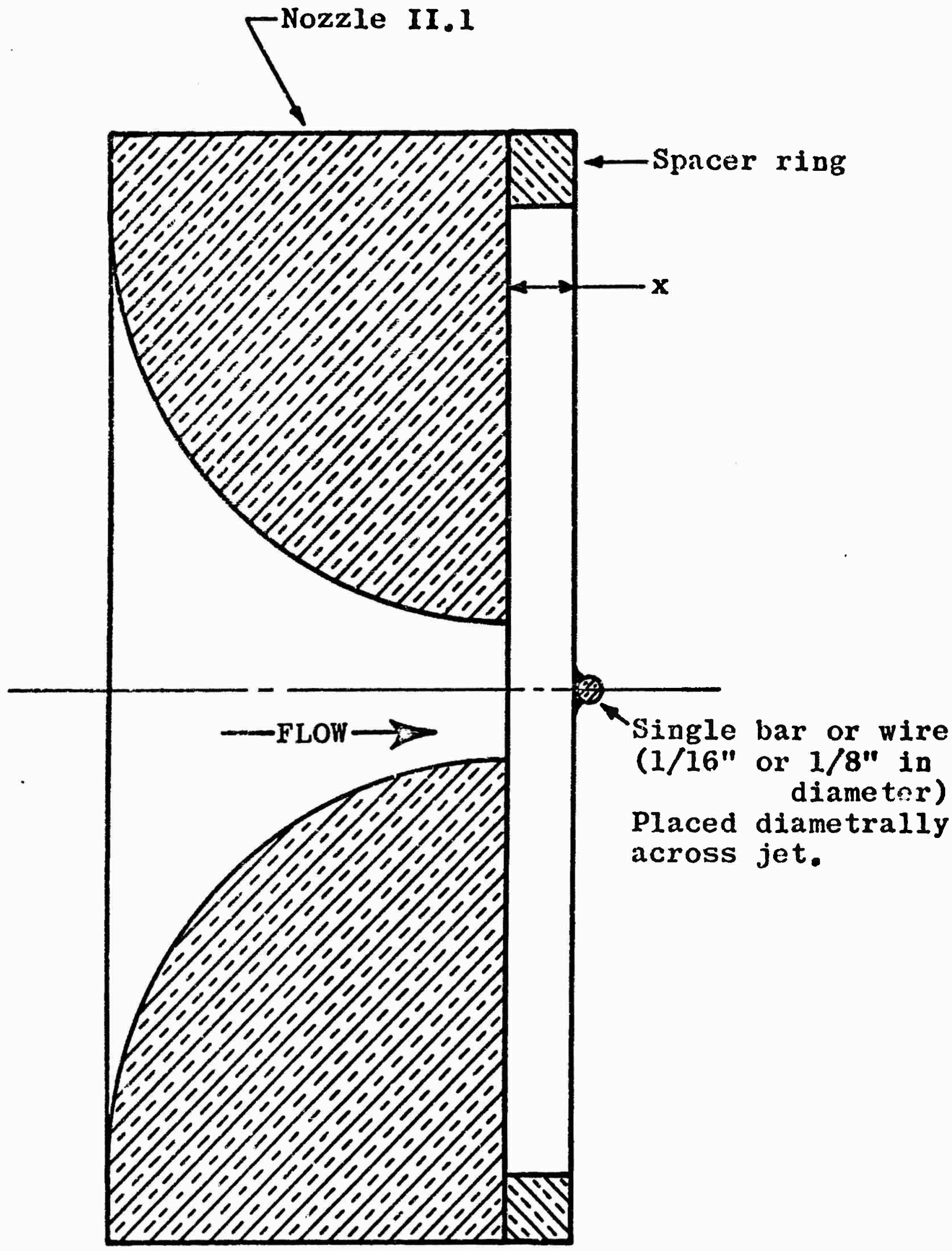


FIGURE II.21. SINGLE BAR OR WIRE ACROSS JET.

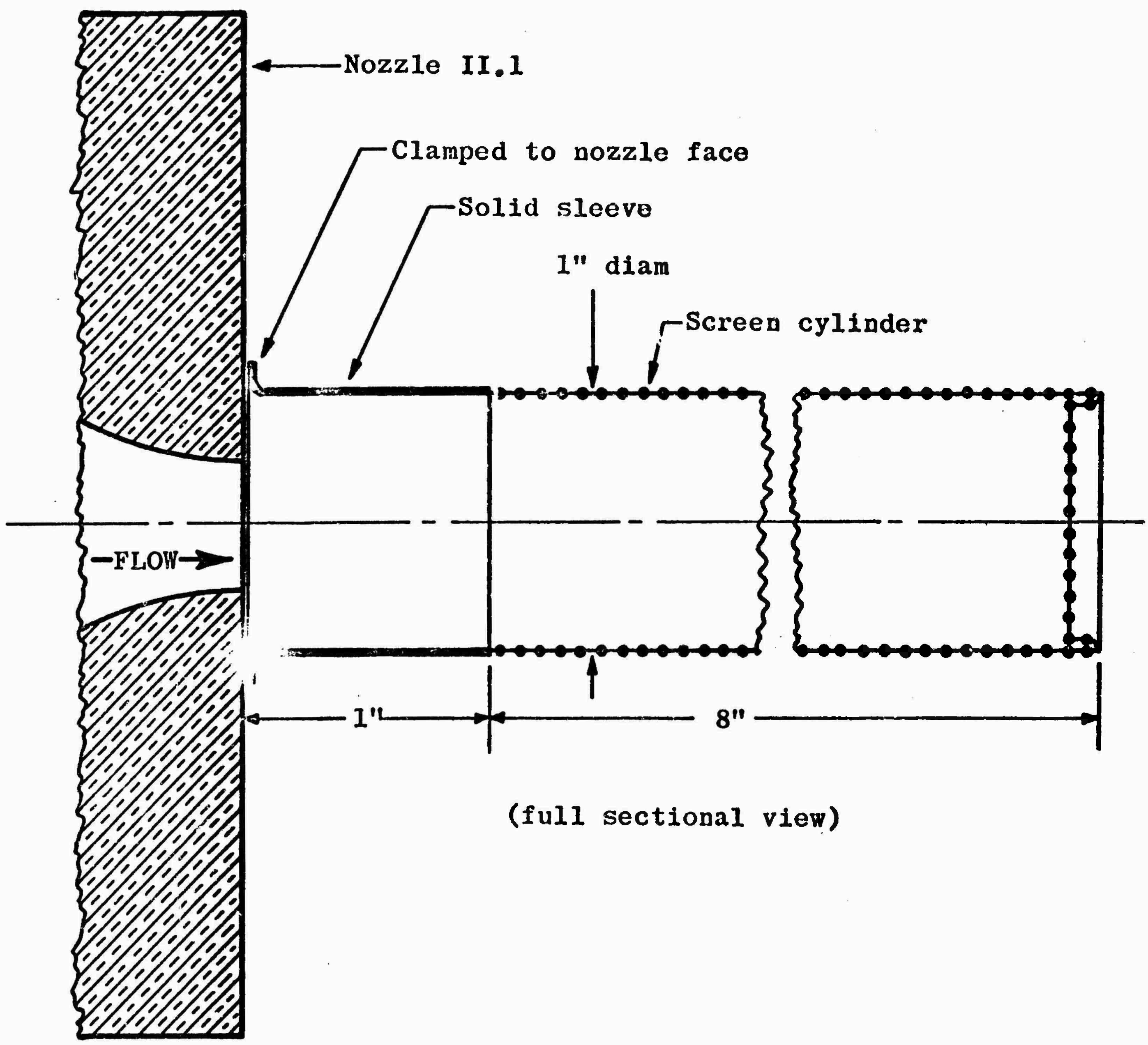


FIGURE II.22. SCREEN CYLINDER.

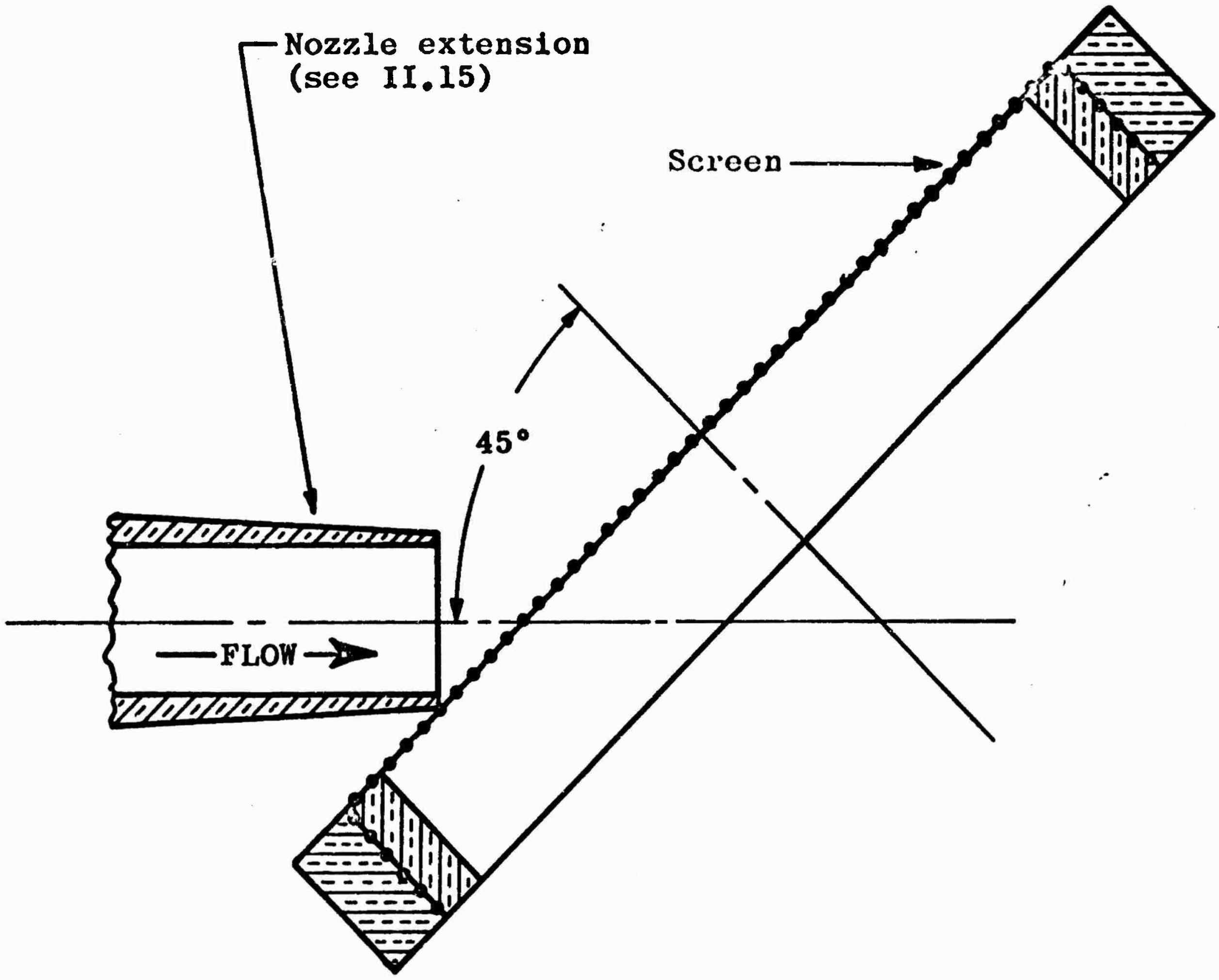


FIGURE II.23. SCREEN AT 45 DEGREES TO JET AXIS

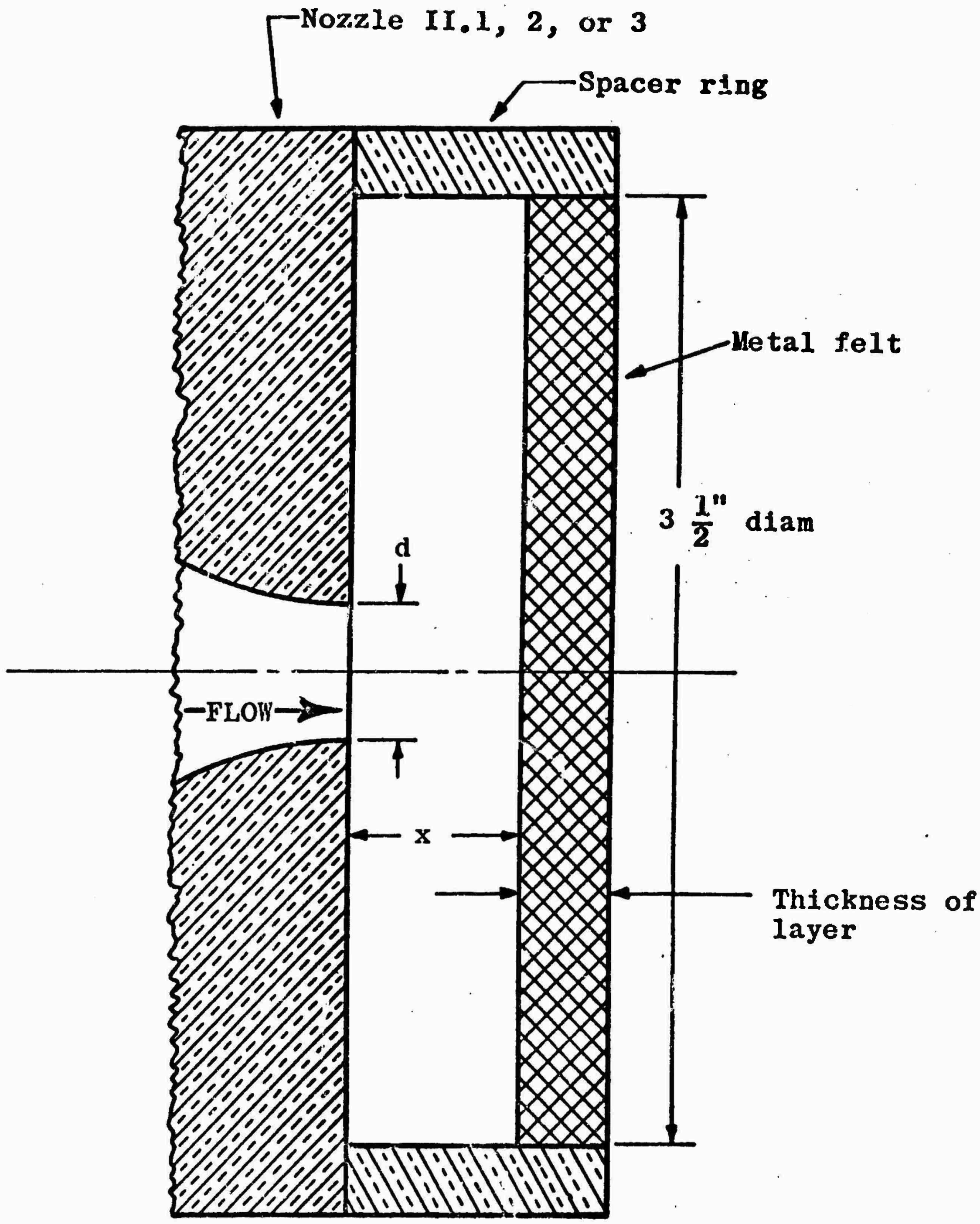
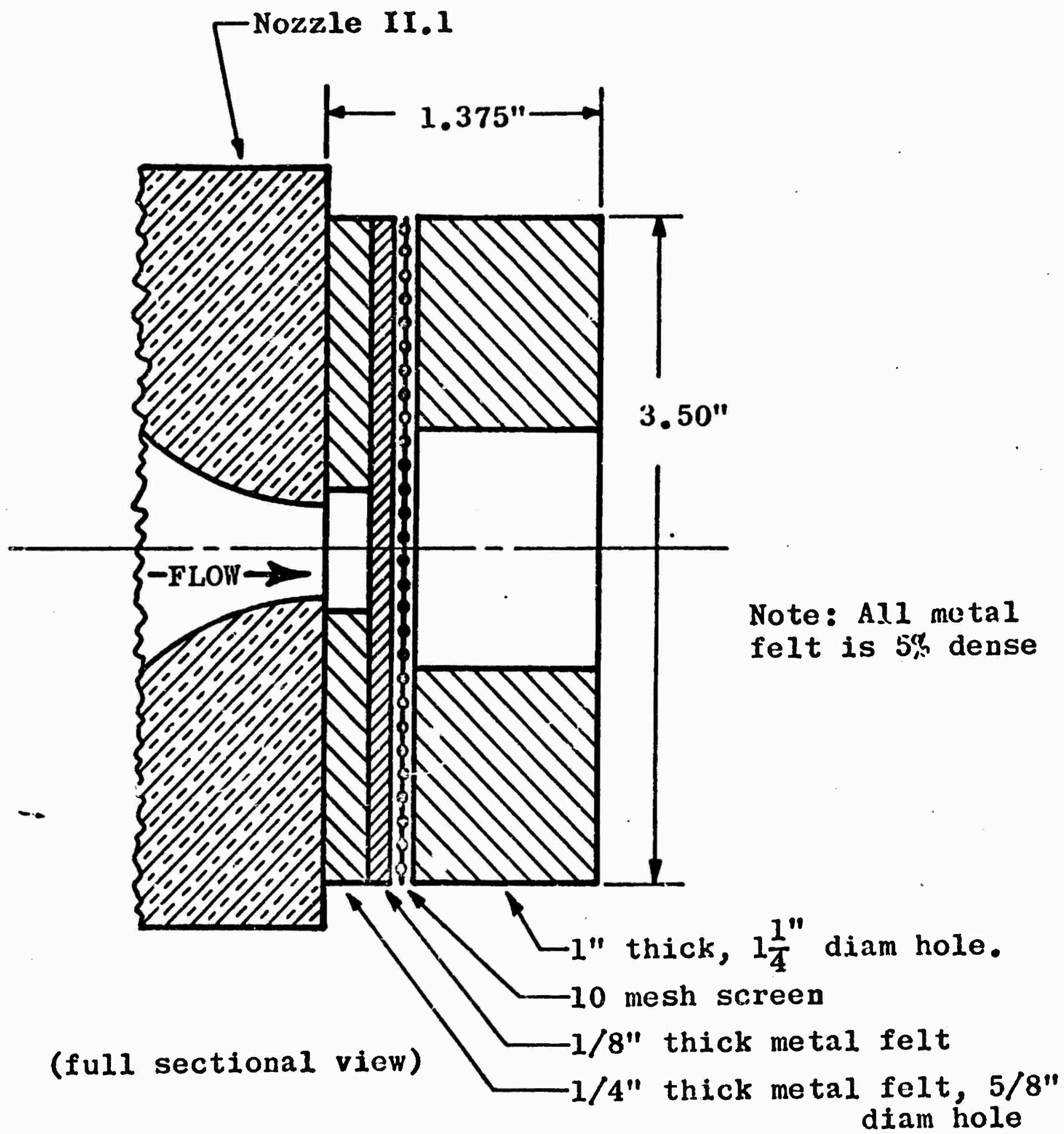


FIGURE II.24. METAL FELT.

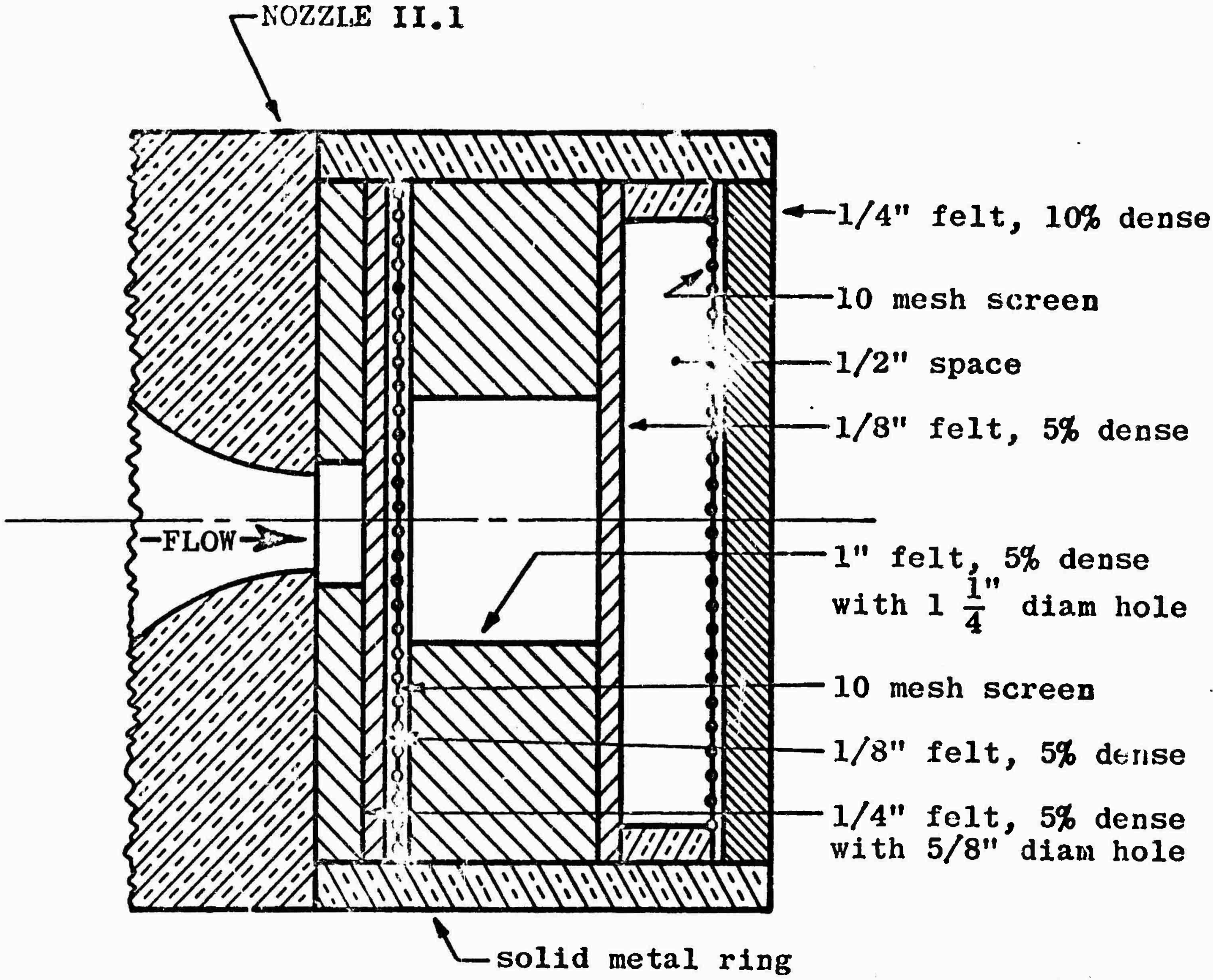


II.25A - As shown in sketch.

II.25B - As shown in A except enclosed within a solid metal ring.

II.25C - Like B except for addition of a terminal 1/8" thick layer of metal felt supported by a 10 mesh screen.

FIGURE II.25. METAL FELT SILENCER WITHOUT SOLID BOUNDARIES.



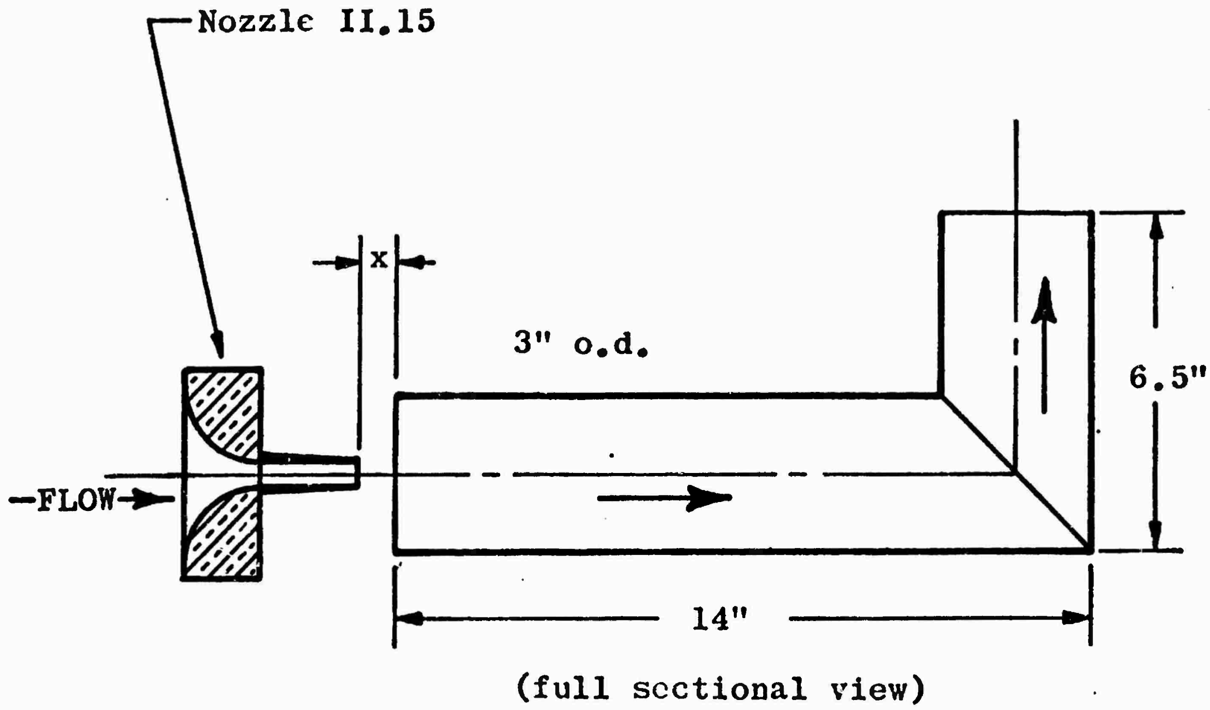
(full sectional view)

II.26A - As shown in sketch.

II.26B - As shown in A plus an additional terminal layer of 1/8" thick, 20% dense felt.

II.26C - As in B plus an additional terminal layer of 1/16" thick, 20% dense felt.

FIGURE II.26. METAL FELT SILENCER WITH SOLID BOUNDARIES.

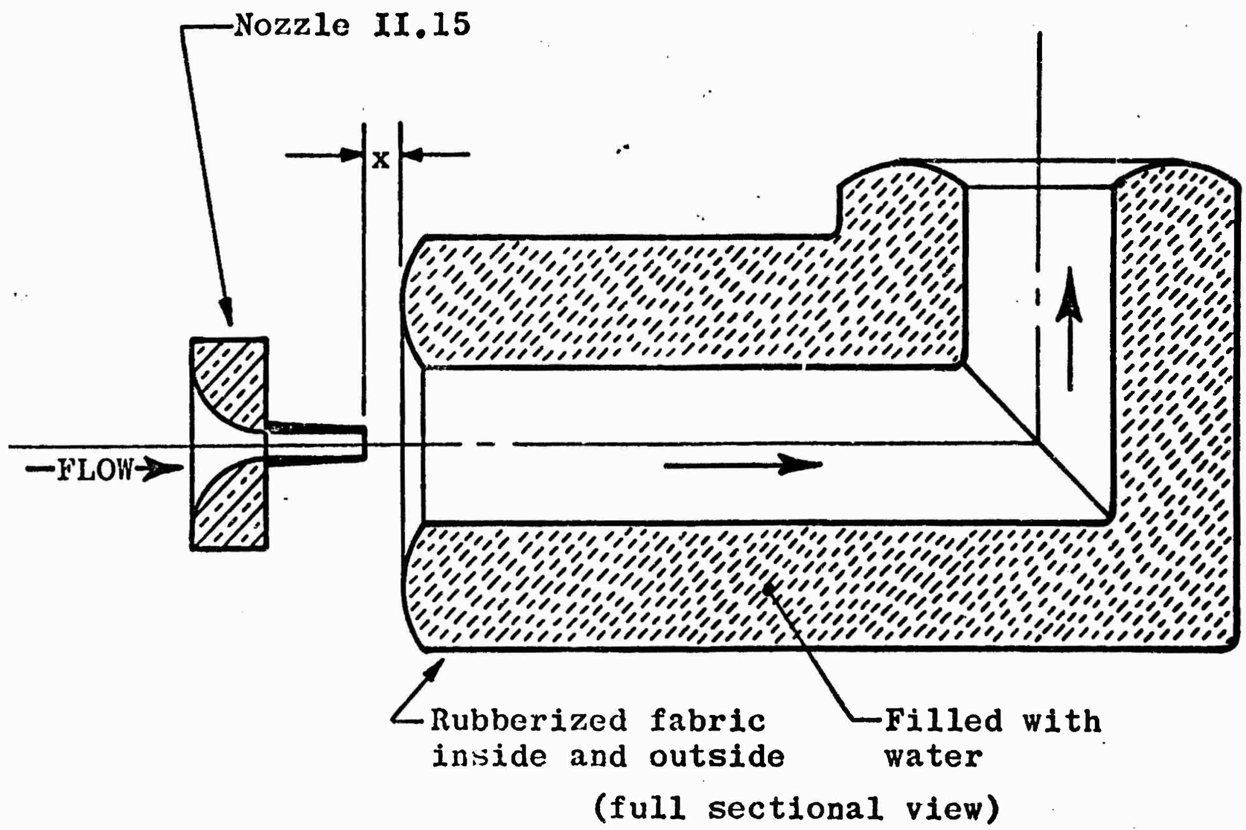


II.27A - As shown in sketch. Muffler body fabricated from 3" o.d. stainless steel tubing, wall thickness 0.062".

II.27B - As shown in sketch except fabricated from perforated metal plate rolled into tubing. Perforated plate has 65% solid area; 1/8" diam holes, 0.185" oc in triangular pattern.

II.27C - Same as B except wrapped on the outside with 3/8" thick layer of fine fiberglass having thin neoprene coating outermost.

FIGURE II.27. SOLID AND PERFORATED MUFFLER BODIES.



Detailed internal structure of the water-filled part not known. The tubular passages for the air flow have practically the same dimensions as muffler body II.27A.

FIGURE II.28. WATER-INFLATED FLEXIBLE MUFFLER BODY.